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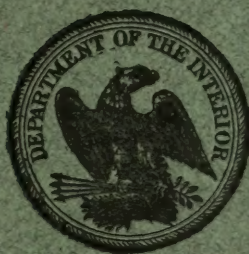
BUREAU OF MINES

JOSEPH A. HOLMES, DIRECTOR

THE TRANSMISSION OF HEAT INTO
STEAM BOILERS

BY

HENRY KREISINGER AND WALTER T. RAY



WASHINGTON

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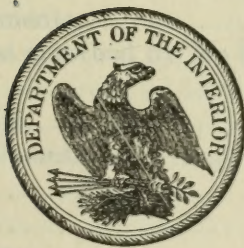
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THE TRANSMISSION OF HEAT INTO STEAM BOILERS.

By HENRY KREISINGER and WALTER T. RAY.

INTRODUCTION.

The investigation of the transmission of heat into steam boilers is one of several researches now being carried on by the Bureau of Mines that have for their object the testing of methods by which the mineral fuels in this country may be used more efficiently. A better understanding of the laws governing heat transmission into boilers will result in the design of more efficient boilers, and more efficient boilers will reduce waste in the use of fuel.

The results of the investigations described in this report indicate that the conductivity of the heating plates of steam boilers is so high that the present steaming capacities can be tripled or quadrupled by forcing over the heating surfaces three or four times the weight of gases now passed over them. With well-designed mechanical-draft apparatus this greater weight of gases can be forced through the boilers at a small operating cost. It is possible to increase the capacity of many of the present boilers in this way without reducing their efficiency much; in fact by a proper arrangement of the heating surfaces the efficiency can be made higher than the present rating. The efficiency of any boiler can be increased by arranging its heating surfaces in series with respect to the path of hot gases. New boilers of high efficiency can be constructed by making the cross section of the gas passages small in comparison with the length.

In Bryan Donkin's book, entitled "The Practical Physics of the Modern Steam Boiler," it is stated that James Watt obtained nearly as good an evaporation per pound of coal and per square foot of heating surface as is obtained now. The remark is probably true. At any rate, about the only decided superiority of the modern steam boilers over older types is in mechanical construction, and for this superiority credit should be given to the designers of machine tools quite as much as to the boiler engineer. The main reason for tardiness in boiler improvement probably lies in the reluctance of educated engineers to do the dirty and disagreeable work involved in boiler tests.

Nearly a hundred years of practical investigation of boiler and furnace problems has resulted in little advance. Perhaps the main reason why many of the investigations failed to bring about progress was that boiler and furnace were considered a unit and were investigated together. Various combinations of boilers and furnaces have been built and tested without thoughtful planning. Many of the published results of such tests confuse the performance of the boiler and the furnace in such a way that it is difficult, if not impossible, to tell which of the two should be blamed or praised for the poor or good results obtained from the combined apparatus. Evidently, many persons have thought that the combined efficiency could be greatly increased by some mysterious manipulation.

The principles governing the combustion of fuel in boiler furnaces and the absorption of heat by boilers have been little understood. The dogmas that the area of grate should have a certain ratio to the area of the heating surface, and that it takes 10 square feet of heating surface to make one boiler horsepower, seemingly had become so thoroughly fixed in the mind that they were hardly ever questioned. It is only within the last decade that a few engineers have broken away from the old rule of thumb methods and have begun to investigate the functions of the boiler and furnace separately. Their studies seem to mark the beginning of advance in steam-generating apparatus.

The boiler is the metallic vessel that contains water and steam and absorbs heat; consequently it should be studied as a heat absorber.

The furnace is that part of the steam-generating apparatus in which the potential energy of the coal is changed into heat; consequently it should be studied as a heat generator.

ACKNOWLEDGMENTS.

The authors wish to express their appreciation of the encouragement given them by their superiors, J. A. Holmes, L. P. Breckenridge, H. M. Wilson, and J. C. Roberts, and of the routine work done in the laboratory and computing room by their assistants, C. E. Augustine, F. E. Woodman, F. J. Bird, G. W. Thompson, and C. J. Fletcher. To J. K. Clement, formerly physicist of the technologic branch of the United States Geological Survey, the authors are indebted for much valuable information regarding the physical principles involved in various phases of heat propagation.

Special thanks are due to H. G. Stott and W. S. Finlay, jr., of the Interborough Rapid Transit Co., of New York, and to A. Bement, of the Commonwealth Edison Co., of Chicago, who kindly furnished the data of their work as presented at the end of this bulletin.

PURPOSE OF THIS BULLETIN.

Early in the routine work of the steam-engineering section of the technologic branch of the United States Geological Survey it was realized that boiler tests made for the purpose of determining the value of coal for steam production meant little unless the boiler's performance was subtracted so as to show the efficiency of the grate and furnace. When attempts were made to separate the performance of the boiler from that of the furnace, it was found that little was known of either. This lack of essential knowledge of the fundamental principles of the process of steam production induced the steam-engineering section to take up the study of the heat-absorbing abilities of the heating surfaces of boilers. These investigations were carried on at intervals for nearly three years, during which time many original experiments with small laboratory boilers and with large boilers were made and studied. The purpose of this bulletin is to present the results of these investigations; to discuss the laws that determine the rate of heat transfer from furnace gases through metal plates into boiler water; and to offer suggestions as to how these laws may be applied.

The authors realize, however, that, so far as concerns steam generation, a study of the absorption of heat is only half the duty; the other half is the study of the combustion of the fuel in the furnace. The systematic studies of heat production and absorption being made by the Bureau of Mines treat the two as independent problems. At the mining experiment station of the bureau, at Pittsburgh, Pa., is a furnace with a long combustion chamber. This combustion chamber receives the gases from a fuel bed supplied by a Murphy stoker, and discharges into a steam-boiler setting. The chamber is about 40 feet long and 3 feet square internally. Along one side are large peep-holes 5 feet apart, through which observations can be made and instruments inserted. The space in the combustion chamber is of uniform cross section, that the stream lines of gas flow may not mix except by natural diffusion and unavoidable eddying. In such a furnace the process of combustion can be studied much better than in the ordinary boiler furnace.

SCOPE OF THIS BULLETIN.

The material presented in this bulletin is arranged, for convenience, in five parts, as follows:

- I. Introductory statements.
- II. Description and results of original investigations of the steam-engineering section of the United States Geological Survey.
- III. Abstracts from the works of others and discussion thereof.
- IV. General discussion of the laws governing the three modes of heat propagation.

V. Practical application of the laws governing heat transmission to the design of steam plants and steam boilers and to methods of increasing the capacity and improving the economy of boilers already installed.

SYNOPSIS OF PART I.

Part I contains an exposition of the path of heat travel from the source of the heat to the boiler water. It also contains an explanation of true boiler efficiency and its relation to other efficiencies used in many reports on boiler trials. General deductions are given at the end of this part.

SYNOPSIS OF PART II.

Part II contains the original investigations made by the steam-engineering section of the United States Geological Survey. These investigations consist mainly of about 300 tests with small laboratory multitubular boilers which were fed with air heated in an electric furnace. Laboratory apparatus was used because of the ease with which the running conditions could be controlled. The objects of the tests were:

(1) To find the effect of increasing the velocity of the air on the rate of heat absorption and on the true boiler efficiency, when the temperature of the air entering the boiler remains constant.

(2) To find the effect of raising the temperature of the air entering the boiler on the rate of heat absorption and on the true boiler efficiency, when the initial temperature of the air remains constant.

(3) To find the effect of varying the diameter of the flues on the true boiler efficiency, when the initial velocity and the temperature remain constant.

(4) To find the effect of varying the length of the flues on the true boiler efficiency, when the initial velocity, the temperature of the air, and the diameter of the flues remain constant.

In this part of the report are also discussions of related special observations taken during tests made with large boilers, a torpedo-boat boiler, and a locomotive boiler.

SYNOPSIS OF PART III.

Part III comprises a brief historical review of such work by other investigators as bears directly or indirectly on the subject of heat transmission into boilers, as applied in steam-engineering practice. Abstracts or short quotations are made from those writings that, although almost unnoticed, are of extreme value to engineers. The writings of the following investigators are given particular attention: Osborne Reynolds, John Perry, Dr. Nicolson, and E. Stanton, of England.

SYNOPSIS OF PART IV.

Part IV contains a detailed mathematical discussion of the physical laws governing the three modes of heat transmission. The conclusions arrived at mathematically are tested by the results obtained from the experiments made by the technologic branch of the United States Geological Survey.

SYNOPSIS OF PART V.

Part V treats of the application of the laws of heat transmission to steam boilers, and of the possibilities of improving boilers already installed by special arrangements of baffles and grates. It further gives a few instances where such principles as are developed in Parts II, III, and IV have been successfully applied, and concludes with a brief summary of the significance of surface combustion as applied by Prof. Bone to boiler heating.

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PART I.

PRELIMINARY CONSIDERATIONS.

HOW HEAT FLOWS INTO THE BOILER WATER.

Before any summary of deductions from the investigation can be given and intelligently followed, it is necessary to show briefly how heat travels from the hot fuel bed, the flames, and the hot gases, through the heating plates of the boiler into the boiler water. A few definitions and explanations of the terms used will help to a better understanding of the conclusions. Figure 1 shows diagrammatically

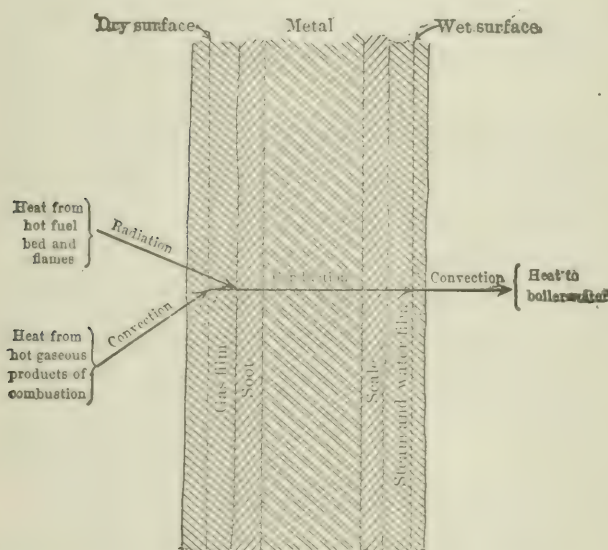


FIGURE 1.—Diagrammatic illustration of the ways in which heat enters, travels through, and leaves a boiler plate.

the modes of heat travel into boiler water.

The diagram shows the metal plate covered with a layer of soot on the gas side and with a layer of scale on the water side. Next to the soot layer and entangled in its recesses is a layer or film of motionless gas. This film of gas under ordinary conditions adheres so tightly to the soot or the metal that it may almost be considered a part of the solid plate. It is reasonable, therefore, to assume that the dry surface of the plate is situated some-

where within this film of gas, as indicated in figure 1. The wet surface of the heating plate is perhaps located on the inside of the boiler in a film of water and steam adhering to the layer of scale, or, if the boiler is clean, to the metal plate.

In any steam-generating apparatus the heat is produced in the furnace by the burning of fuel. As the process of combustion goes on the heat evolved is immediately absorbed, partly by heating of the fresh fuel but mainly by the gaseous products of combustion. The absorption of heat by these substances causes a rise in their temperature, so that by observing the temperature one can tell how much heat they contain.

The heat evolved and contained in the gaseous products of combustion is transferred through the gas-filled space and then transmitted through the heating plates into the boiler water. The process of transmission takes place in three distinct modes or ways (fig. 1), each of which is governed by a definite law not applicable to the others. Before the heat reaches the body of the boiler water it changes its mode of travel at least twice. It is first imparted to the dry surface of the heating plate in two ways: (a) By radiation from the hot fuel bed, the furnace walls, and the luminous flames, and (b) by convection from the hot, moving gaseous products of combustion. When the heat reaches the dry surface it changes its mode of travel and passes through the soot, metal, and scale to the wet surface purely by conduction. From the wet surface of the plate the heat is carried into the boiler water mainly by convection.

It may be well to define and describe briefly these modes of heat travel and the dry and the wet surfaces of the heating plate.

RADIATION.

Radiation is that mode of heat propagation by which heat passes through space from one body to another without any material agency; the two bodies do not come into contact directly nor does a third body or other bodies transmit the heat. In the case of a furnace and the heating plate of a boiler, the heat from the hot fuel bed and furnace walls flows by radiation through the space filled with gases, without heating the gases appreciably, directly into the boiler plate. The quantity of heat transmitted by radiation would not be lessened if the gases did not fill the space; in fact, it would be slightly greater if there were a vacuum between the fuel bed and the plate.

CONVECTION.

Convection of heat from one place to another always implies the motion of a fluid receiving or giving up the heat. Thus, in the case of the hot gases and the boiler, the heat is transferred to the heating plate (from the gas) by small particles of the gas moving from the body of the gas to the plate and imparting their heat to the latter by their impact upon it. In other words, convection is a process of continuous interchange of position between cooled particles (molecules) of gas next to the heat-absorbing surface and hotter particles within the body of the gas. It will be shown later that the quantity of heat imparted to a unit area of dry surface of the plate depends on the rate at which these particles of gas exchange positions.

CONDUCTION.

Conduction of heat is the process by which heat flows from a hotter body to a colder one, when the two bodies are in contact, or from a hotter to a colder part of the same body. Conduction implies no visible nor mechanical motion of different bodies or different

parts of the same body, but implies a direct contact. In a boiler the heat passes by conduction from the particles of hot gas that touch the soot coating, into the soot, and then through the metal and the scale (if there happens to be any of the latter) into the particles of water next to the scale. These particles of water absorb heat until they change into small bubbles of steam which are carried away by the circulation of the water. Since the heat in these steam bubbles is taken from the heating plate through the water into the steam space by mechanical, or visible motion, it is said to travel by convection.

The particles of gas in contact with the soot coating of the tubes may form a gas film of appreciable thickness; then the heat near the outside of the film has to travel by conduction through part of the thickness of the film a longer distance than does the heat near to the soot coating. Thus the boundary along which convection ceases and conduction starts is not a well-defined surface, but rather a gas layer of measurable thickness. In addition, gases are permeable to radiation, so that the heat traveling by radiation penetrates the gas film and enters the soot coating almost directly. In consequence, the dry surface of the heating plate may be defined as the thin layer of gas near, or at, the outside surface of the soot coating, where the heat ceases to travel by convection and starts to travel by conduction. For similar reasons the wet surface of the heating plate may be defined as the thin layer of steam and water, near or at the surface of the layer of scale away from the metal plate, where the heat ceases to travel by conduction and starts to travel by convection facilitated by the circulation of the water in the boiler.

HEAT AVAILABLE FOR A BOILER AND TRUE BOILER EFFICIENCY.

Without the aid of an external agency heat flows from a hot body only to bodies at lower temperatures; therefore a boiler can absorb only that heat which is above the temperature of the water in it. The heat that is above the temperature of the boiler water is called the "heat available for the boiler." Heat below this temperature will not flow into the boiler and therefore is not available for it. The quantity of heat a boiler will absorb per unit of time depends almost entirely on the quantity of available heat delivered to it. Any commercial boiler absorbs only part of the available heat. The heat absorbed, expressed as a percentage of the heat available, is termed the "true boiler efficiency." This efficiency depends somewhat on the way the heat is presented to the boiler, but chiefly upon the arrangement of the boiler's heating surface. The true boiler efficiency is thus a ratio that is expressed by the equation:

$$\text{True boiler efficiency} = \frac{\text{Heat absorbed by boiler}}{\text{Heat available for boiler}}$$

The true boiler efficiency has been devised because it is the only true measure of the boiler's ability to absorb heat. All other boiler efficiencies used in commercial boiler tests blame the boiler for not absorbing heat that is below the temperature of the boiler water and therefore not available for absorption.

It is perhaps well to stop here and explain more fully by specific examples what the true boiler efficiency is and how it is related to the boiler efficiency ordinarily used.

Assume that 1 pound of a combustible having a heating value of 12,000 British thermal units is burned completely and that the gase-

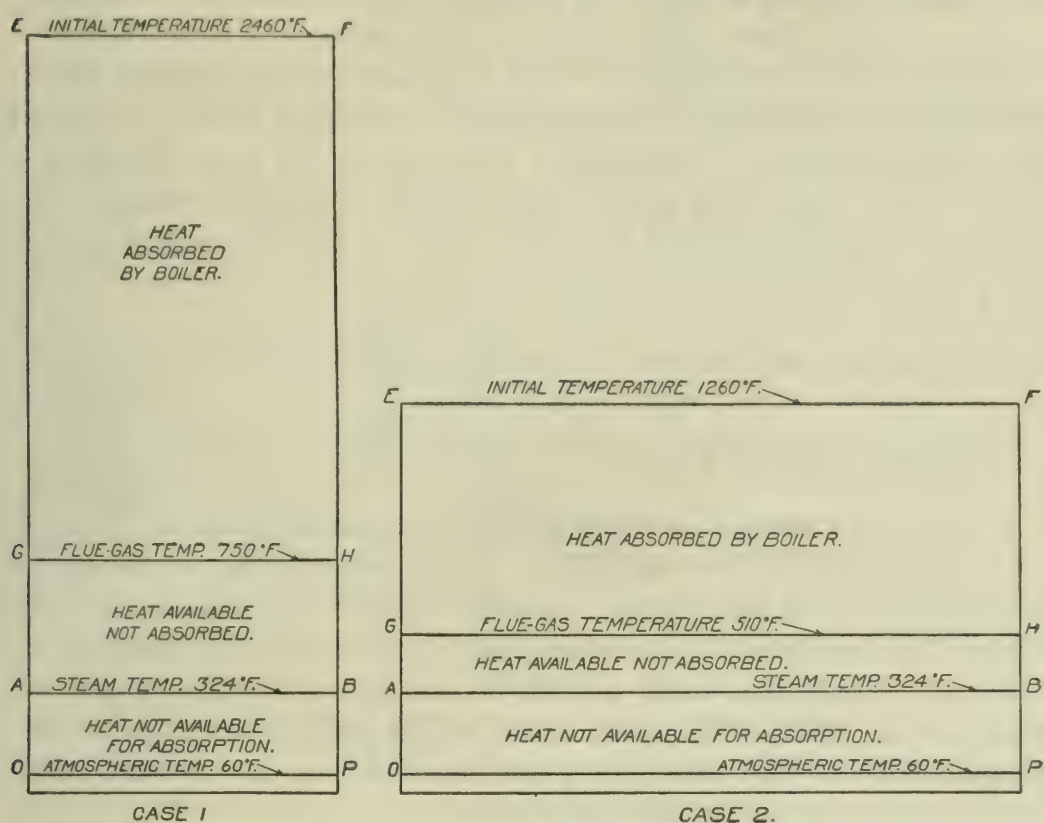


FIGURE 2.—Heat available for absorption by a boiler under two different temperature conditions. Case 1 shows the various quantities of heat when 1 pound of "combustible" is burned with so much air that the resulting gaseous products of combustion weigh 20 pounds and have an initial temperature of 2,460° F. Case 2 shows the various quantities of heat when 1 pound of "combustible" is burned with so much air that the resulting gaseous products of combustion weigh 40 pounds and have an initial temperature of 1,260° F.

ous products of combustion weigh 20 pounds; assume also, for the sake of simplicity, that the specific heat of the gases is 0.25; then the temperature of the products of combustion will be 2,400° F. above the temperature of the atmosphere. If the latter temperature is assumed to be 60° F., the temperature of the products of combustion is 2,460° F. Then if, as in figure 2, case 1, the results are platted, the temperature being used as ordinates and the product of the weight of the gases multiplied by their specific heat as abscissæ, the heat generated by the burning of the combustible is represented by the area OEFP, the line OP representing the atmospheric temperature.

Assume that this hot gas is passed through a boiler generating steam. If the pressure of the steam were 80 pounds, the temperature of the boiler water would be about 324° F. Let the line AB indicate this temperature. Then of the total heat generated, only the heat above 324° F., or the heat represented by the area AEFB, is available for absorption. The remainder of the heat represented by the area OABP, being below the temperature of the water in the boiler, will not flow into the water, and therefore is not available for absorption. Only the available heat represented by AEFB can be absorbed by an ideal boiler; therefore this heat is taken as the basis for measuring the heat-absorbing ability of a boiler.

Now, assume that the boiler cools the gases down to 750° F. If this temperature is indicated in the diagram by the line GH, the heat absorbed by the boiler is represented by the area GEFH. Then the true boiler efficiency is shown by the ratio of the area GEFH to the area AEFB; and the ordinarily used boiler efficiency is shown by the ratio of the area GEFH to the area OEFP. The numerical values of the two efficiencies in the assumed cases are:

$$\text{The true boiler efficiency, } E_4 = \frac{2,460 - 750}{2,460 - 324} = 80 \text{ per cent.}$$

$$\text{Efficiency ordinarily used, } E = \frac{2,460 - 750}{2,460 - 60} = 71 \text{ per cent.}$$

If the steam, instead of being generated under 80 pounds of pressure, were generated under a pressure of 200 pounds, the corresponding temperature of the boiler water would be 388° F. On account of this higher boiler-water temperature, less heat would be available for absorption and less heat would be absorbed by the boiler, even though the ratios of the two heats might remain the same. If the temperature of the escaping gases assumed in the second case be 800° F., the two boiler efficiencies are:

$$\text{True boiler efficiency, } E_4 = \frac{2,460 - 800}{2,460 - 388} = 80 \text{ per cent.}$$

$$\text{Efficiency ordinarily used, } E = \frac{2,460 - 800}{2,460 - 60} = 69 \text{ per cent.}$$

This second case shows that, although the rise in steam pressure may not affect the true boiler efficiency, it lowers the ordinarily used boiler efficiency.

Again, assume that the same 1 pound of combustible is burned completely and that the resulting weight of the gaseous products of combustion is 40 pounds. The temperature of the products of combustion will then be only $1,200^{\circ}$ F. above atmospheric temperature. The various quantities of heat are represented in figure 2, case 2. Although the quantity of heat generated is the same as that in the first case,

twice as much of the heat is below the temperature of the boiler water, and, therefore, not available for absorption; the quantity of the heat available is smaller by the same amount. Thus, even should the true boiler efficiency remain constant, a smaller percentage of the total quantity of heat generated will be absorbed because less heat is available for absorption. If the assumed temperature of the gases leaving the boiler be 510° F., the two efficiencies are:

$$\text{True boiler efficiency, } E_4 = \frac{1,260 - 510}{1,260 - 324} = 80 \text{ per cent.}$$

$$\text{Efficiency ordinarily used, } E = \frac{1,260 - 510}{1,260 - 60} = 62.5 \text{ per cent.}$$

The above figures show that, although the true boiler efficiency is the same, the ordinarily used efficiency is 8.5 per cent lower than that in case 1 (fig. 2). Finally, so much air could be used for combustion that the temperature of the products would equal that of the boiler water; then no heat would be available for the boiler, and no heat would be absorbed by it. Thus it can be seen that the useful effect of the heat generated diminishes with the drop in its initial temperature. Any factor that reduces the temperature of the products of combustion is the fault of the furnace and not of the boiler. It is therefore wrong to charge, in the above cases, the decreases in the useful effect to the boiler; they are the fault of the furnace and should therefore be charged against the furnace.

True boiler efficiency has the advantage over the ordinarily used boiler efficiency that it takes care of the variation of the temperature of the furnace gases as well as the variation of the temperature of the boiler water, caused by different steam pressures. In other words, true boiler efficiency does not blame the boiler for a lessened steam production caused by a lower temperature of the furnace gases, which really is a fault of the furnace, nor does it blame the boiler for absorbing less heat when the temperature of the boiler water is raised by raising the steam pressure.

GENERAL DEDUCTIONS.^a

In almost all the boilers now used for generating steam only a small percentage of the heating surface is so exposed to radiation from the fuel bed, furnace walls, and flames as to receive heat, both by radiation and convection. By far the greater part of the surface receives heat only by convection from the moving gaseous products of combustion.

The greatest resistance to the flow of heat is met before the hot gases reach the dry surface of the heating plate. If a boiler is even

^a These deductions are placed in the first part of the bulletin for the convenience of the reader although the justifications for some of them appear later.

moderately clean, the resistance of the plate itself to the flow of heat through it is very small indeed. The resistance to the passage of the heat from the plate into the boiler water is also very small. It may be said that the heating plates transmit to the boiler water all the heat that can be imparted to their dry surfaces.

Increasing the rate at which heat is imparted to the dry surface of the heating plate increases the rate of steam production in the same proportion.

Since in most boilers the heat imparted by convection is the larger part of the total heat the boiler receives, increasing the rate at which heat is imparted to the heating surface by convection causes a nearly proportionate increase in the rate of steam production. If the initial temperature of the moving gases remains constant, increasing the velocity with which the latter pass over the heating plate increases in an almost direct ratio the rate at which heat is imparted to the dry surface of the plate, and therefore increases almost directly the rate at which steam is made.

In the future the velocity at which gases pass over the heating plates will be the factor that will be used for increasing the capacity of boilers. There is no valid reason why stationary boilers can not be worked at two or three times the rate at which they are worked at present.

To increase the capacity of any boiler, pass more gases over its heating surfaces.

A boiler that has its heating plates arranged in such a way that the gas passages are long and of small cross section is more efficient than a boiler in which the gas passages are short and of large cross section.

If the same weights of gas at the same initial temperature are passed through a 2-inch and a 4-inch flue, both flues having the same length, the 2-inch flue will absorb more heat than the 4-inch flue, although the 4-inch flue has twice as much heating surface as the 2-inch flue.

Locomotive boilers are as a rule more efficient than stationary fire-tube boilers, because the tubes in the former boilers have a smaller diameter in comparison with their length than the tubes in the latter boilers.

To increase the efficiency of water-tube boilers, insert baffles in such a way that the heating surfaces are arranged in series with reference to the gas flow, thus making the gas passage longer.

As the velocity of gases over the heating plates of a boiler increases, the true boiler efficiency at first drops, and then, after the gases exceed a certain velocity, it remains nearly constant.

In multitubular boilers the true boiler efficiency increases as the diameter of the tubes decreases. In water-tube boilers the true

boiler efficiency increases as the cross section of the gas passages between individual tubes decreases.

In multitubular boilers the true boiler efficiency increases as the length of the flues increases, and in water-tube boilers it increases as the length of the gas passages increases.

In boilers that receive the greater part of their heat by convection, the true boiler efficiency after the gases attain a certain velocity is nearly independent of the initial temperature of the gases.

Within certain limits, the true boiler efficiency depends on the shape of a boiler and not on its size nor its operation.

PART II.

INVESTIGATIONS OF THE UNITED STATES GEOLOGICAL SURVEY.

EXPERIMENTS WITH SMALL MULTITUBULAR BOILERS.

Laboratory experiments with small multitubular boilers were undertaken by the technologic branch of the United States Geological Survey with the object of obtaining some data that could be used to study the factors influencing the rate at which heat is imparted by convection to the heating surfaces of boilers.

DESCRIPTION OF APPARATUS.

The apparatus used in these experiments consisted mainly of small horizontal interchangeable tubular boilers which were fed with air heated in an electric furnace. The steam generated in a boiler was condensed in a small surface condenser and the quantity accurately determined. A steam ejector, connected to the exit end of the boiler, caused a current of air to flow through the boiler and furnace, which were placed within a long box that formed the body of the apparatus. The box was made of "asbestos slate," and the annular space around the boiler was filled with mineral wool and magnesia covering. The current consumed in the furnace was controlled by a water rheostat and was measured by an integrating wattmeter. Figure 3 shows the apparatus as assembled.

THE BOILERS.

Experiments have been made with four horizontal multitubular boilers. A piece of standard 4-inch wrought-iron pipe served for the outside shell of each boiler. The heads were made of $\frac{1}{8}$ -inch sheet copper and were riveted to the ends of the 4-inch pipe. All boilers had ten flues of thin copper tubing; these were expanded and soldered into holes in the boiler heads. Figure 3 shows two sections through one of the boilers. The following table gives the dimensions of the four boilers.

TABLE 1.—*Dimensions of small experimental boilers.*

Dimensions.	Boiler No.—			
	1	2	3	4
Length of boiler, outside to outside of boiler heads.....inches..	8.28	8.28	16.125	8.28
Actual outside diameter of flues.....do.....	.252	.313	.252	.3745
Actual inside diameter of flues.....do.....	.175	.230	.175	.2928
Thickness of walls of flues.....do.....	.038	.041	.038	.0408
Total area of inside cross section of flues.....square feet..	.00167	.002885	.00167	.00467
Total heating surface of flues.....do.....	.31617	.41554	.6156	.529

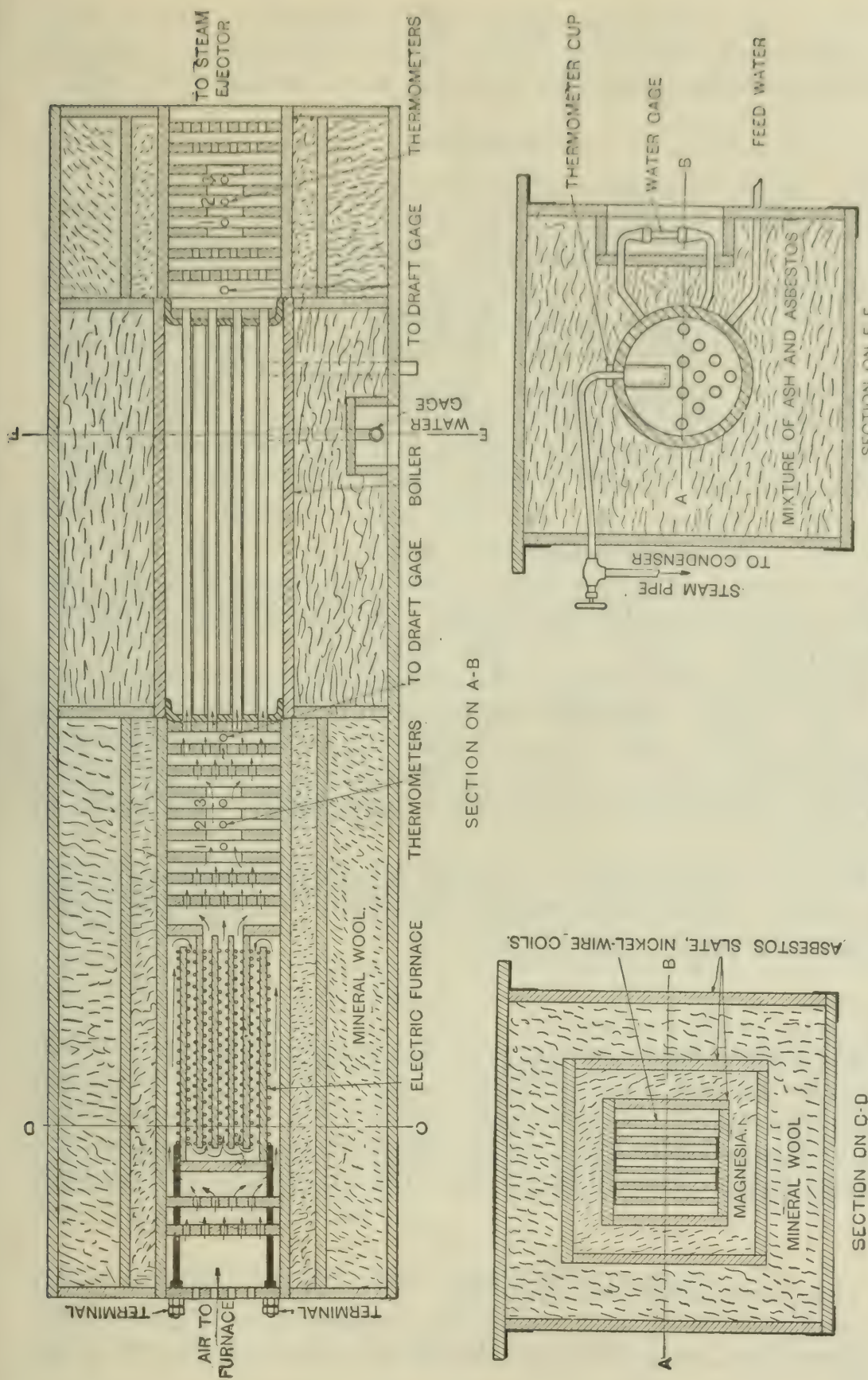


FIGURE 3.—Apparatus for studying the absorption of heat in small multitubular boilers.

Each of the boilers was equipped with a water-gage glass, a thermometer cup for measuring the temperature of the steam, a pipe for feeding the boiler, and a pipe for letting out the steam.

THE FURNACE.

The electric furnace consisted of six coils of pure nickel wire wound around rectangular pieces of asbestos slate, $\frac{9}{32}$ inch thick, $3\frac{7}{8}$ inches wide, and 12 inches long. The nickel wire was 0.072 inch in diameter, about No. 13 B. & S. gage. There were 51 turns in each coil, so that the total length of the wire in the furnace was 204 feet. In most of the tests the coils were connected in series; with high temperatures and high-pressure drops or "drafts"^a three coils were connected in series, and the two threes were connected in parallel. A water rheostat was used for adjusting the electrical energy to each temperature and "draft." Direct current of 220 volts was used for feeding the furnace. The coils were placed vertically in the front portion of the asbestos-slate box, with a space of about $\frac{5}{16}$ inch between them, and were so baffled that the air passed along the coils three times before it left the furnace. The arrangement of the coils in the furnace is shown in two views in figure 3.

Endurance of the furnace.—With temperatures up to about 900° F. there was no particular trouble, but when the temperature of air was raised to 1,200° F. and 1,500° F. the nickel wire seemed to crystallize and become very brittle after cooling. The coils usually failed by fusing after they had been in use ten hours a day for six to ten days.

It must be understood that in order to raise the temperature of air to 1,500° F. with extremely high or low "drafts" the coils were heated close to the point of fusion (2,600° F.).

THE CONDENSER.

The surface condenser consisted of two concentric copper tubes, the inner one of which was for steam and the outer one for a stream of cooling water. The size of the steam tube was the same as that of the flues of boiler No. 1; the internal diameter of the water tube was 0.3 inch. The total length of the condenser was 11 feet 6 inches.

THE EJECTOR.

The steam ejector for producing "draft" was a standard $\frac{1}{4}$ -inch pipe; it was placed in the center of a contracted pipe, made of galvanized sheet iron, which took the hot air away from the rear of the boiler.

^a Throughout this bulletin the word "draft" is inclosed in quotation marks if used, or rather misused, in the ordinary sense of a pressure difference between two points in a moving stream of gas.

THE BODY OF THE APPARATUS.

The "asbestos-slate" box that formed the body of the apparatus was 1 foot high by 1 foot wide by 5 feet long. The space that held the furnace and the space back of the boiler were 4 inches square, so that the total thickness of the insulating walls was 4 inches. The walls around the furnace consisted of a $\frac{1}{4}$ -inch lining of asbestos slate, and a 1-inch layer of magnesia boiler covering, composed of 85 per cent magnesium oxide and 15 per cent asbestos. The space between the layer of magnesia and the outside sheet of the box was filled with mineral wool. The space around the boiler was filled with a mixture of wood ashes and asbestos.

The "asbestos slate" proved to be easily workable; it withstood the temperature of the furnace very well. Asbestos "shingles" were found to be more durable at high temperatures than asbestos "lumber," as they are called by the maker.

PROTECTING SCREENS.

The space between the furnace and the boiler was fitted with eight perforated screens. The four middle screens had in their centers round holes $1\frac{1}{4}$ inches in diameter. In the spaces between these four screens three thermometers were so inserted that their bulbs were in the center line of the $1\frac{1}{4}$ -inch holes. The object of these screens was to partly confine the air around the bulbs of the thermometers in order that its temperature could be more accurately measured. The two screens nearest to the furnace and the two nearest to the boiler each had thirty-two $\frac{1}{4}$ -inch holes so drilled that the thermometers were perfectly screened from direct radiation from either the boiler or the furnace. A similar arrangement was employed for obtaining the temperature of the air leaving the boiler. Tubes for measuring "drafts" were inserted on both sides of the boiler between the boiler head and the first screen. The front of the furnace was also screened from radiating heat to the outside. Both of the boiler heads were covered with $\frac{1}{4}$ -inch asbestos paper and slate, so that only the heating surface of the flues was effective. In figure 3 the path of the air through the apparatus is indicated by the arrowheads. The boiler was fed with distilled water obtained by condensing steam from the main boiler of the plant.

METHOD OF CONDUCTING EXPERIMENTS.

Before starting a test the apparatus was brought to the temperature at which the test was to be run. After a test was started all conditions were kept as nearly uniform as possible. The duration of each test was from 30 minutes to two hours. With high pressure drops ("drafts") and high temperatures the tests were shorter,

because enough water was evaporated in the shorter time to make the tests as accurate as a two-hour test with the low pressure drops and low temperatures.

DETERMINATION OF THE HEAT ABSORBED BY THE BOILERS.

The heat absorbed by the boilers was calculated from the water evaporated, which was condensed and accurately measured. No attempt was ever made to correct for moisture in the steam.

TEMPERATURES.

Temperatures up to 700° F. were measured with mercurial thermometers, which were compared with a standard thermometer calibrated by the Bureau of Standards in Washington. Temperatures above 700° F. were measured by exposed platinum and platinum-rhodium thermocouples which were calibrated by means of the known melting points of pure metallic silver, barium nitrate, potassium chloride, and calcium chloride; the point of 700° F. was determined by direct comparison with a mercurial thermometer immersed in an oil bath.

Although great care was taken in placing and screening the thermometers and the thermocouples, considerable difficulty was experienced in measuring the initial and final temperatures of the air. With the initial temperatures of $1,200^{\circ}$ F. and $1,500^{\circ}$ F., the three thermocouples would at times read as much as 100° F. apart, and for no apparent reason. Generally the thermocouple closest to the furnace read higher than the other two. The only reasonable explanation of this fact was that in spite of the screens this thermocouple received some heat by radiation from the furnace, whereas the one nearest to the boiler lost some heat by radiation to the boiler. The readings also differed considerably if the thermometers or thermocouples were slightly displaced. The thermometers on the side of the leaving gas read from 10° F. to 20° F. apart. It was difficult to come to any conclusion as to which of the thermometers or thermocouples were right and which were wrong, and it was thought best to take the average of the three readings at either end. The many difficulties met in measuring the temperature made the men conducting the experiments realize that measuring the temperature of permanent gases requires great skill.

EXTENT OF THE EXPERIMENTS.

Three to five series of tests were run with each boiler. The initial temperatures were nearly the same in all the tests of each series. Each series consisted of 15 to 30 tests, which were run with different pressure drops. Thus the tests in each series show how the velocity

of the air passing along the heating surface affects the rate of heat absorption; the different series of tests with the same boiler show the effect of the initial temperature; and the tests with different boilers give the influence of the diameter and the length of the flues.

DATA TAKEN DURING EXPERIMENTS.

In all the tests made the following temperature readings, on the ordinary Fahrenheit scale, were taken.

t_a = temperature of air entering furnace.

T = temperature of air entering boiler (initial temperature).

t_f = temperature of air leaving boiler.

t_s = temperature of steam.

t_w = temperature of feed water.

Also the following:

d = pressure of air entering boiler, measured in inches of water.

D = pressure of air leaving boiler, measured in inches of water.

W = water condensed measured in cubic centimeters and reduced to pounds for calculations.

E = electrical energy, in watt-hours, consumed during test.

METHOD OF CALCULATION.

The weight of air,^a M , passing through the boiler per second is obtained from the equation:

$$W[965.7 + (t_s - t_w)] = 0.2375M(T - t_f)$$

$$\text{Hence } M = \frac{W[965.7 + (t_s - t_w)]}{0.2375(T - t_f)}$$

The volume, V , of the air entering the boiler is determined from the relation based on the fundamental properties of permanent gases, that

$$\frac{V}{12.387M} = \frac{T + 461}{493}$$

$$\text{Hence } V = \frac{(T + 461) \times 12.387M}{493}$$

and the velocity, v , of the air entering the boiler is

$$v = \frac{V \times 144}{nd^2 \times 0.7854}$$

where d is the internal diameter expressed in inches, and n the number of flues in the boiler.

^a In all calculations no account was taken of the moisture in the atmosphere.

If, in the calculation above indicated, W is reduced to water evaporated per second, then the velocity, v , is in feet per second.

The heat absorbed by the boiler per second is given by the equation

$$H = W[965.7 + (t_s - t_w)]$$

where W is expressed in pounds of water evaporated per second.

The true boiler efficiency, E_4 , is given by the ratio

$$\frac{\text{Heat absorbed by boiler}}{\text{Heat available for boiler}}$$

and is computed by the equation

$$E_4 = \frac{T - t_f}{T - t_s}$$

The true boiler efficiency was also computed from the heat supplied by the electrical energy. However, the efficiency so calculated was less consistent than that calculated in the manner outlined above. This was probably on account of the variation in the quantity of heat lost by radiation.

RESULTS OF EXPERIMENTS.

Tables 2, 3, 4, and 5 give the data and results of all reliable tests made on the small boilers described. In these tables the figures at the tops of the temperature columns indicate the positions of the thermometers as shown in figure 3.

TABLE 2.—Data and results of tests on boiler No. 1.

No. of test.	Duration of test (minutes).	"Draft" (inches water).			Temperatures (° F.).										Water condensed (c. c.).	Heat absorbed per second (B. t. u.).	Air passed through boiler per second (pounds).	Velocity of air entering boiler per second (feet).	True boiler efficiency (E.).	Heating surface per boiler horsepower (square feet).
		Entering boiler.	Leaving boiler.	Difference.	Air entering boiler.			Air leaving boiler.				Air entering furnace.	Steam.	Feed water.						
					Thermometers—			Average.	Thermometers—											
					1	2	3		1	2	3									
59	90	0.0	0.3	0.3	506	502		504	59	209	68	222	225		223	152	0.0688	0.00103	15.0	42.5
84A	180	1.1	.6	.5	499	506		502	63	209	62	247			247	490	.1114	.00184	26.6	26.3
84	180	1.1	.6	.5	494	502		498	59	209	67	244			244	502	.1138	.00188	27.2	26.1
60	42	1.1	.8	.7	504	502		503	66	210		236	240		238	150	1.1454	.00251	33.5	26.1
62	54	1.1	1.1	1.0	502	505		504	76	210		251	256		253	214	1.1416	.00238	34.6	20.1
61	45	1.1	1.1	1.0	498	502		500	73	210		254	258		256	392	1.1652	.00285	41.2	17.1
63	42	1.1	1.6	1.5	496	502		499	66	210		263	267		265	246	.2073	.00375	54.1	14.1
71	180	2.2	2.2	2.0	494	503		498	65	210	68	261	263		262	1,063	.2403	.00429	61.9	12.0
72	120	2.2	2.2	2.0	495	505		500	69	211	76	266	268		267	724	.2440	.00441	63.8	12.0
73	120	2.2	2.2	2.0	489	500		494	74	211	89	266	269		268	751	.2502	.00479	66.7	11.7
64	39	3.3	3.3	3.0	496	500		498	73	210		256	261		259		.2720	.00479	69.2	10.8
74A	120	4.4	4.3	3.9	493	503		496	75	209	81	265	268		267	925	.3103	.00665	81.6	9.4
74C	110	4.4	4.3	3.9	491	501		496	74	209	74	264	266		265	939	.3458	.00631	90.9	6.7
74B	120	4.4	4.4	4.0	501	495		498	74	209	73	264	266		265	1,068	.3609	.00654	94.4	8.1
76	120	8.8	8.7	7.9	487	498		492	75	209	78	263	264		264	1,311	.4410	.00812	116.5	6.6
75A	120	9.9	8.8	7.9	481	495		489	71	209	76	260	262		261	1,319	.4576	.00846	120.9	6.4
75B	120	8.8	8.9	8.1	485	499		492	76	209	81	262	264		263	1,248	.4310	.00792	113.5	6.8
65	45	1.3	14.9	13.6	486	504		495	71	210	78	259	262		262	1,440	.5630	.01017	146.4	5.9
77	120	1.7	18.6	16.9	483	503		493	74	209	80	260	262		261	1,775	.5939	.01083	155.5	4.9
85	180	0.0	.5	.5	700	702		701	62	209	61	270			270	854	.1945	.00190	33.3	15.0
85A	180	0.0	.5	.5	700	702		701	62	209	60	266			266	909	.2072	.00201	33.1	14.1
78A	120	1.1	1.1	1.0	702	702		702	75	209	86	289	294		292	848	.2832	.00290	50.9	10.3
79B	80	1.1	1.1	1.0	695	699		697	73	209		294	297		296	668	.2957	.00311	54.1	9.9
80	120	2.2	2.2	2.0	694	704		699	73	209	94	308	312		310	1,299	.4438	.00481	83.9	6.6
80D	100	2.2	2.2	2.0	690	700		695	73	210	80	312	315		314	1,054	.4375	.00483	84.0	6.7
80A	120	2.2	2.2	2.0	690	700		695	76	210	118	311	314		312	1,296	.4382	.00489	83.2	6.8
80B	120	2.2	2.2	2.0	690	700		695	75	210	114	312	315		313	1,306	.4387	.00497	86.8	6.1
80C	120	2.2	2.2	2.0	691	702		695	72	210	81	313	315		314	1,295	.4472	.00518	90.8	6.5
81C	60	4.4	4.4	4.0	680	698		689	72	210	144	317	319		318	722	.4565	.00623	90.6	4.7
81A	120	4.4	4.4	4.0	687	702		695	81	209	86	313	314		314	1,691	.5646	.00625	108.7	3.7

TABLE 2.—Data and results of tests on boiler No. 1—Continued.

No. of test.	Duration of test (minutes).	"Draft," (inches of water).			Temperatures (°F.).										Electrical energy consumed (watt-hours).	Water condensed (c. c.).	Heat absorbed per second (B. t. u.).	Air passed through boiler per second (pounds).	Velocity of air entering boiler per second (feet).	True boiler efficiency (E _b).	Heating surface per boiler horsepower (square feet).	
		Entering boiler.	Leaving boiler.	Difference.	Air entering boiler.			Air leaving boiler.				Air entering furnace.	Steam.	Feed water.								
					Thermometers—			Thermometers—														
					Average.			Average.														
					1	2	3	1	2	3												
81B	120	4	4.4	4.0	683	700		677	318	321	320	76	211	116	2,968	1,624	.5274	.00625	107.1	76.79	5.5	
82	120	8	8.7	7.9	684	701		693	313		314	72	210	85	4,154	2,435	.8138	.00911	156.9	78.32	3.6	
82B	120	8	8.7	7.9	685	706		696	315		315	72	210	92	4,186	2,222	.7379	.00817	142.1	78.37	4.0	
82A	120	8	8.8	8.0	684	707		697	315		315	76	210	100	4,158	2,262	.7456	.00822	143.3	78.45	3.9	
83D	90	8	9.1	8.3	685	707		696	310		310	54	210	58	3,136	1,597	.7293	.00795	138.4	79.53	4.0	
83C	120	1.0	11.2	10.2	682	704		693	314		314	66	211	69	4,534	2,328	.7895	.00876	152.2	78.65	3.7	
83B	120	1.2	13.6	12.4	682	702		692	315		315	66	211	86	5,068	2,644	.8829	.00985	171.0	78.40	3.3	
83A	120	1.3	15.1	13.8	682	705		694	317		317	71	211	85	5,306	2,866	.9579	.01071	186.1	77.94	3.1	
86A	175	0	.5	.5	1,005	990	950	982	302	290	286	293	62	210	62	1,910	1,430	.3346	.00204	44.4	89.29	8.7
86B	155	0	.5	.5	980	975	970	975	300	290	287	292	60	210	60	1,812	1,121	.2970	.00183	39.5	89.26	9.9
88A	180	1	1.1	1.0	990	980	970	980	335	326	322	328	71	210	75	2,776	1,966	.4421	.00286	62.0	84.67	6.6
88B	180	1	1.1	1.0	980	990	970	980	347	335	332	338	67	210	65	3,160	2,147	.4872	.00320	69.3	83.38	6.0
89	120	2	2.2	2.0	1,015	995	985	998	380	367	365	370	74	210	110	2,892	1,796	.6453	.00433	95.0	79.70	4.5
89A	120	2	2.2	2.0	1,000	980	985	988	368	358	356	361	76	210	85	2,866	2,039	.6815	.00458	99.8	80.59	4.3
90	125	4	4.4	4.0	1,005	975	985	988	378	373	371	374	74	210	100	4,462	2,955	.9351	.00641	139.9	78.92	3.1
90A	150	4	4.4	4.0	1,005	970	970	982	380	371	369	373	71	210	88	5,206	3,757	1.0020	.00693	150.5	78.89	2.9
91A	180	8	8.7	7.9	1,005	980	975	987	382	357	354	364	80	210	88	8,340	6,802	1.5110	.01022	222.6	80.18	1.9
91	150	8	8.8	8.0	1,015	970	960	982	389	375	373	379	64	210	84	7,178	4,904	1.3120	.00916	199.0	78.11	2.2
92A	180	1.1	14.6	13.5	985	960	950	965	380	359	348	362	88	210	97	10,436	7,582	1.6710	.01167	250.5	79.87	1.8
92B	180	1.2	15.2	14.0	1,005	975	965	982	385	362	350	366	83	210	100	10,766	8,084	1.7760	.01213	263.5	79.84	1.7

TABLE 3.—Data and results of tests on boiler No. 2.

No. of test.	Temperatures (°F.).										Electrical energy consumed (watt-hours).	Water condensed (c. c.).	Heat absorbed per second (B. t. u.).	Air passed through boiler per second (pound).	Velocity of air entering boiler per second (feet).	True boiler efficiency (E.).	Heating surface per boiler horsepower (square feet).	
	"Draft" (inches of water).		Air entering boiler.				Air leaving boiler.											Feed water.
			Thermometers—		Average.	Thermometers—		Average.										
	Entering boiler.	Leaving boiler.	Difference.	1		2	3		1	2								3
	Duration of test (minutes).	1	2	3	1	2	3	1	2	3								Air entering furnace.
193	120	0.0	0.1	510	497	478	495	247	242	239	243	86	280	161	0.0538	0.00090	7.5	71.50
192	120	.0	.3	496	476	470	481	277	272	268	272	81	652	260	.0872	.00176	14.4	44.10
191	120	.1	.4	519	496	485	500	287	280	280	282	96	788	437	.1445	.00279	23.4	26.00
198	120	.1	.6	504	484	460	483	291	288	285	288	95	914	466	.1541	.00333	25.0	25.00
228	95	.1	.7	512	488	485	495	295	292	289	292	76	880	318	.1350	.00280	23.3	28.50
199	120	.1	.9	512	484	485	495	304	297	294	298	97	1,180	616	.2034	.00467	36.4	18.90
200	120	.1	1.1	515	485	488	503	304	301	302	293	97	1,344	646	.2133	.00447	37.5	18.00
202	95	.2	1.7	513	488	490	497	301	299	296	297	78	1,386	610	.2500	.00546	45.6	14.90
201	75	.2	1.7	511	480	486	492	306	302	301	303	97	1,062	532	.2813	.00627	52.0	13.70
194	120	.3	2.3	507	484	489	493	303	299	289	297	98	1,980	890	.2639	.00631	52.5	13.10
202	120	.5	3.4	505	476	488	490	304	301	299	305	82	2,268	938	.3136	.00714	59.1	12.30
203	95	.7	4.7	515	480	492	496	308	305	305	306	83	2,394	938	.3960	.00873	75.0	9.50
204	95	.8	5.7	508	478	486	491	303	300	296	300	85	2,200	900	.3794	.00826	69.4	10.10
205	95	1.0	7.0	500	469	478	482	302	299	294	298	86	2,356	967	.4073	.00652	76.6	9.40
206	95	1.1	8.0	504	475	485	488	304	302	297	301	85	2,754	1,150	.4835	.01089	90.1	8.00
224	95	1.9	12.1	511	489	493	498	306	304	298	303	77	3,250	1,100	.4920	.01029	88.6	7.80
225	70	2.2	14.0	508	487	491	495	305	304	298	302	78	2,640	835	.4802	.01048	87.3	8.00
207	95	2.3	14.2	510	474	473	486	299	297	291	296	83	2,894	1,230	.5194	.01181	96.0	7.40
226	70	2.6	16.2	505	482	489	492	303	300	295	300	79	2,894	1,003	.5770	.01295	105.1	6.70
232	70	2.5	16.3	511	495	503	503	307	308	302	306	80	2,910	900	.5531	.01182	99.3	7.00
195A	45	2.9	18.6	516	483	488	496	308	306	302	305	94	1,932	708	.6256	.01379	115.0	6.20
195	95	2.9	18.9	509	503	514	509	312	314	304	310	96	4,190	1,632	.6819	.01443	122.0	3.00
231	70	3.0	19.5	515	499	503	506	310	309	303	307	81	3,290	1,010	.8813	.01280	168.6	6.00
196	95	3.2	22.2	501	495	502	499	308	305	299	304	97	4,420	1,598	.6617	.01429	119.5	5.80
218	70	3.5	23.5	510	487	486	494	299	297	291	296	92	3,258	997	.8673	.01296	99.9	4.80
230	70	3.5	23.7	513	494	501	503	308	306	299	306	77	3,554	1,057	.6074	.01298	109.4	6.30
229	65	3.8	25.7	520	504	510	511	316	315	308	313	74	3,546	1,293	.7920	.01284	142.6	4.20
189	120	.0	.2	694	678	663	678	299	296	291	295	96	836	557	.1842	.00262	20.1	20.90
188	120	.0	.3	695	679	660	681	316	303	306	308	95	1,000	585	.1236	.00219	21.7	12.90
190	120	.1	.4	691	670	660	674	330	328	320	326	95	1,228	795	.2533	.00319	31.5	14.00
208	95	.1	.6	680	659	636	658	330	326	321	326	84	1,112	566	.2399	.00300	22.8	16.20
209	95	.1	.9	692	662	650	668	347	349	344	347	86	1,880	810	.3300	.00445	43.8	11.40
210	95	.2	1.2	694	661	652	669	359	355	353	356	87	1,598	918	.3842	.00517	50.9	10.00

TABLE 3.—Data and results of tests on boiler No. 2—Continued.

No. of test.	Duration of test (minutes).	"Draft" (inches of water).			Temperatures (°F.).										Electrical energy consumed (watt-hours).	Water condensed (c. c.).	Heat absorbed per second (B. t. u.).	Air passed through boiler per second (pound).	Velocity of air entering boiler per second (feet).	True boiler efficiency (E _b).	Heating surface per boiler horsepower (square feet).		
					Air entering boiler.				Air leaving boiler.														
		Entering boiler.		Leaving boiler.		Difference.		Thermometers—			Thermometers—			Average.									
		1	2	3	1	2	3	1	2	3	1	2	3	Average.									
																						1	2
211	95	2	1.7	1.5	694	658	652	668	366	363	361	363	363	87	1,842	1,145	4793	.00662	65.0	66.49	8.00		
212	95	.3	2.3	2.3	695	659	655	670	368	365	363	365	363	85	2,238	1,293	5407	.00746	73.6	66.42	7.10		
214	95	.6	4.6	4.0	701	672	670	681	375	371	367	371	367	210	3,200	1,718	7295	.00992	98.7	65.90	6.30		
215	70	.9	6.8	5.9	685	659	658	667	372	369	365	369	369	82	2,674	1,390	7975	.01127	110.9	65.40	4.80		
217	55	1.6	9.5	7.9	698	673	670	680	370	374	369	371	371	83	2,522	1,305	9540	.01300	129.1	65.90	4.10		
216	70	1.5	9.5	8.0	696	676	662	678	374	372	364	370	370	83	3,174	1,610	9255	.01235	125.5	66.10	4.20		
220	70	2.2	14.2	12.0	694	685	670	683	375	372	366	371	371	89	3,830	1,860	1,0620	.01433	142.9	66.10	3.60		
221	70	2.7	18.6	15.9	700	677	674	684	380	378	378	378	378	212	4,310	2,080	1,1860	.01638	163.4	64.63	3.20		
222	70	3.6	24.0	20.4	692	669	667	676	377	375	367	373	373	87	2,187	1,265	1,2650	.01758	174.2	66.10	3.00		
223	70	4.1	28.1	24.0	693	670	668	677	378	377	369	373	373	86	3,094	2,378	1,3970	.01880	193.1	64.95	2.80		
184	120	0	1	1	905	885	865	885	325	328	320	324	324	90	1,372	785	2621	.00197	23.1	83.20	14.70		
177	125	.1	.3	.2	910	905	885	900	344	330	330	330	330	92	1,368	878	2791	.00206	24.4	82.60	13.80		
241	120	0	.3	.3	970	925	895	930	441	433	429	435	435	93	686	505	2864	.00204	26.6	82.20	13.40		
124	110	.1	.6	.5	980	960	930	957	441	433	431	435	435	86	1,808	1,042	3728	.00301	37.1	76.78	10.30		
123	120	.1	.6	.5	912	894	863	890	409	397	402	403	403	87	1,816	1,133	3815	.00380	38.8	75.76	10.10		
234	70	.1	.7	.6	885	885	885	885	405	400	384	396	396	90	1,288	850	4845	.00417	48.9	72.62	7.90		
235	70	.1	.8	.7	885	885	885	885	415	411	397	406	406	91	1,490	908	5179	.00455	53.4	70.94	7.40		
156A	120	.2	1.0	.8	932	910	871	904	429	424	418	424	424	90	2,940	1,700	5650	.00496	59.0	69.50	6.80		
122	120	.3	1.3	1.0	930	903	868	900	441	436	432	436	436	89	2,700	1,550	5205	.00472	56.0	70.52	7.40		
236	70	.2	1.2	1.0	905	895	900	900	435	429	417	427	427	91	1,622	1,093	6225	.00555	55.5	68.61	6.20		
183	95	.3	1.5	1.2	895	895	880	890	426	424	424	425	425	91	2,230	1,612	6770	.00613	72.2	68.45	5.70		
233	70	.2	1.7	1.5	895	875	885	885	443	440	426	436	436	89	2,018	1,325	7560	.00709	83.2	66.60	5.10		
185	95	.3	2.0	1.7	935	920	875	910	426	425	421	424	424	93	2,592	1,673	7002	.00607	72.5	69.08	5.50		
237	100	.5	2.4	1.9	933	937	924	938	444	438	435	439	439	84	3,466	1,975	7834	.00661	80.6	69.00	4.90		
237	70	.4	2.4	2.0	885	865	875	875	434	429	416	425	425	87	2,366	1,400	7999	.00748	87.1	67.79	4.80		
250	70	.6	3.6	3.0	890	906	904	900	434	428	419	427	427	81	2,814	1,680	8083	.00833	98.7	68.63	4.10		
238	70	.5	3.6	3.1	910	880	895	895	452	447	437	445	445	93	2,856	1,766	8042	.00842	111.2	66.10	3.80		
125	85	.9	4.7	3.8	924	918	890	911	484	479	478	480	480	72	3,628	2,024	9820	.00960	114.7	68.00	3.90		
251	70	.8	4.8	4.0	900	910	905	905	444	440	429	438	438	84	3,184	1,940	1,1700	.01060	126.2	67.35	3.50		
120	110	1.0	5.1	4.1	967	940	900	936	482	478	486	488	488	88	5,434	3,024	1,1070	.01027	125.0	65.99	3.50		
248	60	1.2	7.0	5.8	895	920	915	910	432	439	427	433	433	90	3,200	1,890	1,2520	.01105	132.0	68.40	3.10		
240	70	.9	6.8	5.9	915	875	895	895	480	456	446	454	454	96	3,714	2,310	1,3100	.01250	147.6	65.03	2.90		
240A	65	1.2	9.2	8.0	865	850	865	860	454	449	438	447	447	95	3,780	2,315	1,4100	.01439	165.6	63.90	2.80		

249	60	1.5	9.6	8.1	880	890	885	885	439	439	426	435	91	213	100	3,654	2,117	1.3940	.01304	155.0	66.85	2.80
252	60	1.9	11.8	9.9	910	925	910	915	454	448	436	440	85	214	91	4,014	2,321	1.5440	.01385	106.2	66.91	2.50
242	45	1.8	12.0	10.1	910	880	880	890	453	446	435	445	85	214	90	3,060	1,655	1.4760	.01388	163.4	65.88	2.50
244	60	1.8	11.9	10.1	910	900	880	890	440	435	423	432	91	213	92	3,904	2,220	1.4760	.01356	139.7	67.60	2.60
243	45	2.2	13.9	11.7	905	855	865	875	453	446	434	444	89	215	94	3,078	1,778	1.5710	.01534	178.6	65.31	2.50
247	60	2.8	17.9	15.1	910	920	903	911	453	446	433	444	90	216	97	4,652	2,610	1.7280	.01559	186.4	67.75	2.20
261	70	1	2	1	1,275	1,245	1,245	1,245	412	411	394	406	83	211	113	1,152	916	.5110	.00257	38.1	51.14	1.50
187	120	1	2	1	1,235	1,182	1,182	1,183	437	430	415	399	79	210	78	2,168	1,430	.4825	.00259	37.1	80.70	1.60
173	120	1	3	2	1,235	1,206	1,185	1,210	437	432	415	426	82	211	82	1,442	1,065	.5775	.00310	45.2	78.50	1.60
256	70	1	3	2	1,245	1,205	1,205	1,207	454	374	437	374	80	210	82	1,804	1,292	.4323	.00219	31.9	83.55	1.60
256	70	1	6	5	1,225	1,175	1,175	1,215	462	455	437	451	80	211	90	1,860	1,173	.6060	.00363	33.8	76.02	1.60
161	120	2	7	5	1,222	1,201	1,201	1,199	492	482	475	483	80	210	83	1,136	1,888	.6251	.00368	63.2	72.40	1.60
106	95	2	9	7	1,231	1,195	1,195	1,215	499	491	489	493	83	211	84	2,424	1,690	.7073	.00476	68.8	71.43	1.60
258	70	3	1	1	1,250	1,215	1,180	1,215	521	514	505	513	83	211	118	2,988	2,228	.9038	.00557	84.9	69.95	1.60
167	95	2	1	1	1,236	1,196	1,168	1,203	513	507	506	509	84	212	91	3,310	2,303	.9657	.00518	80.6	69.50	1.60
182	95	3	1	1	1,228	1,208	1,173	1,203	515	507	505	509	89	212	91	3,278	2,390	.9832	.00612	88.4	68.91	1.60
183	95	3	1	1	1,239	1,191	1,159	1,196	521	516	516	516	85	212	111	3,786	2,603	.9657	.00655	96.4	69.72	1.60
181	95	4	2	1	1,244	1,205	1,160	1,203	517	513	507	512	85	212	91	3,786	2,603	1.0100	.00655	96.4	69.72	1.60
229	60	5	2	1	1,210	1,220	1,200	1,210	554	550	543	549	85	213	117	2,792	2,033	1.0100	.00655	96.4	69.72	1.60
169	95	4	2	1	1,228	1,179	1,140	1,182	522	518	514	518	83	212	105	3,758	2,750	1.1400	.00723	103.5	68.45	1.60
300	30	8	4	2	1,225	1,230	1,210	1,225	593	578	578	583	76	215	79	1,986	1,296	1.7400	.01144	108.2	63.64	2.20
200	60	9	4	2	1,225	1,230	1,210	1,225	593	578	578	583	76	215	79	1,986	1,296	1.7400	.01144	108.2	63.64	2.20
171	95	7	4	2	1,220	1,176	1,133	1,176	525	521	520	522	86	214	87	3,536	2,422	1.4950	.00965	137.3	67.88	2.60
289	95	8	5	4	1,240	1,230	1,198	1,193	526	523	517	522	95	216	65	3,346	2,070	1.5850	.01518	143.4	68.55	2.40
288	45	1.4	7.8	6.4	1,245	1,240	1,225	1,237	595	595	587	592	64	217	88	3,936	2,422	2.0070	.01496	208.2	63.24	1.90
281	60	1.9	10.2	8.3	1,225	1,240	1,205	1,223	593	592	581	589	73	218	112	3,562	2,552	2.3300	.01553	228.0	63.09	1.60
283	60	2.3	12.8	10.5	1,230	1,250	1,220	1,233	608	606	595	603	73	221	99	6,112	3,857	2.5520	.01688	249.1	62.30	1.60
284	60	2.9	17.2	14.3	1,255	1,260	1,250	1,255	617	616	604	612	74	221	98	6,462	4,181	2.7700	.01813	271.2	62.20	1.40
285	45	3.3	19.8	16.5	1,235	1,255	1,230	1,247	623	621	608	616	74	222	105	7,170	4,470	2.9400	.01962	292.0	61.07	1.20
286	45	3.8	23.4	19.6	1,235	1,238	1,230	1,234	621	622	600	614	72	224	105	5,770	3,500	3.1900	.02160	317.6	61.89	1.20
293	45	4.3	27.5	23.2	1,245	1,240	1,245	1,237	627	625	616	621	71	227	104	6,302	3,737	3.2850	.02248	333.0	61.00	1.20
287	45	4.4	28.0	23.6	1,265	1,255	1,240	1,247	635	631	616	626	72	226	98	6,698	3,757	3.3200	.02251	335.0	61.80	1.20
306	45	1	4	3	1,525	1,550	1,505	1,527	464	452	446	454	84	211	85	1,138	832	.7420	.00202	50.4	81.58	1.10
305	45	1	6	5	1,535	1,565	1,520	1,540	527	512	503	514	82	211	87	1,356	1,045	.9290	.00383	60.3	75.28	1.10
304	45	2	1.0	8	1,520	1,550	1,500	1,523	584	568	562	571	78	212	84	1,782	1,265	1.0730	.00476	82.2	73.28	1.10
303	50	3	1.3	1.0	1,495	1,520	1,480	1,498	595	587	583	588	74	213	82	2,170	1,321	1.2280	.00568	91.8	70.82	1.10
302	45	4	1.5	1.5	1,530	1,555	1,510	1,532	632	613	609	618	72	213	96	2,419	1,638	1.4450	.00696	113.0	69.30	1.10
307	45	4	2.3	1.9	1,485	1,515	1,475	1,492	656	638	637	644	72	214	86	2,606	1,845	1.6450	.00818	125.0	66.22	1.10
296	45	4	3.2	2.6	1,520	1,545	1,508	1,534	666	650	651	656	72	216	84	3,096	2,000	1.7800	.00857	148.8	66.70	1.10
295	45	6	3.2	2.6	1,540	1,555	1,518	1,538	671	655	653	660	72	216	84	3,644	2,409	2.1300	.01064	188.2	66.41	1.10
295	45	8	4.3	3.5	1,510	1,525	1,510	1,512	686	668	669	674	71	218	86	3,772	2,562	2.2850	.01164	197.3	64.80	1.08
297	45	1.0	5.2	4.2	1,500	1,525	1,495	1,512	703	684	686	691	75	221	88	4,382	2,940	2.6200	.01344	231.0	63.62	1.47
298	45	1.4	7.5	6.1	1,515	1,525	1,495	1,512	703	684	686	691	75	221	88	4,382	2,940	2.6200	.01344	231.0	63.62	1.47
299	45	1.6	8.6	7.0	1,505	1,520	1,490	1,505	708	689	691	696	75	223	91	4,762	3,213	2.8850	.01457	254.0	63.15	1.18

TABLE 4.—Data and results of tests on boiler No. 3.

No. of test.	Temperatures (°F.).										Duration of test (minutes).									
	"Draft" (inches of water).			Air entering boiler.			Air leaving boiler.													
				Thermometers—			Thermometers—													
	Entering boiler.	Leaving boiler.	Difference.	1	2	3	Average.	Thermometers—				Average.								
								1	2	3										
276	125	0.00	0.50	0.50	688	683	675	682	232	256	230	231	72	780	413	0.1342	0.00125	21.60	95.50	42.4
265	120	.10	1.10	1.00	683	683	676	681	262	255	258	258	78	1,182	690	.2322	.00232	39.80	88.40	24.5
266	95	.10	1.60	1.50	681	683	669	678	280	273	273	275	74	1,134	720	.3060	.00320	54.80	86.45	18.6
277	120	.20	2.20	2.00	681	681	681	681	286	281	282	283	93	1,628	1,130	.3749	.00397	68.20	84.68	15.2
267	95	.10	2.20	2.10	683	687	682	684	291	285	285	287	83	1,322	890	.3768	.00400	69.20	84.10	15.1
268	70	.30	3.20	2.90	687	691	687	688	292	288	287	289	90	1,212	852	.4862	.00514	88.80	83.80	11.7
278	70	.20	3.30	3.10	682	682	684	684	290	285	285	283	100	1,092	860	.4860	.00512	88.10	84.74	11.7
264	70	.30	4.20	3.90	683	697	680	687	283	278	277	279	84	1,368	947	.5432	.00561	97.00	85.90	10.5
279	70	.20	4.20	4.00	678	675	679	677	281	277	277	278	100	1,280	1,020	.6040	.00637	109.20	85.62	9.4
269	60	.40	5.40	5.00	684	688	684	684	277	276	276	277	91	1,258	970	.6448	.00665	114.90	87.00	8.8
280	70	.30	6.20	5.90	676	677	679	677	273	271	270	271	115	1,520	1,184	.6594	.00684	117.20	87.13	8.6
263	70	.40	6.70	6.30	683	688	684	685	276	272	271	273	84	1,654	1,208	.6930	.00708	122.20	87.62	8.2
262	70	.50	8.30	7.90	681	696	683	687	270	270	272	272	81	1,846	1,358	.7762	.00788	136.20	88.10	7.3
270	60	.60	10.50	9.90	684	685	684	685	272	267	266	268	94	1,716	1,423	.9430	.00953	164.60	88.15	6.0
271	60	.70	12.60	11.90	685	687	687	686	277	269	257	268	88	1,836	1,590	1.0620	.01070	184.70	88.20	5.4
272	60	.80	14.60	13.80	684	686	687	686	279	270	279	273	87	1,978	1,740	1.1510	.01174	205.40	87.18	5.0
273	60	.90	16.90	16.00	682	682	684	682	277	271	270	273	104	2,096	1,878	1.2340	.01271	218.70	87.00	4.6
274	60	1.00	19.10	18.10	684	686	687	686	279	273	271	274	102	2,214	1,970	1.2940	.01322	228.40	86.90	4.4
275	55	1.10	20.80	19.70	683	685	686	683	275	274	280	275	97	2,108	1,970	1.4210	.01467	252.70	86.80	4.0
308	100	.00	1.00	1.00	885	885	885	885	269	263	262	265	80	1,060	810	.3263	.00222	45.00	91.95	17.4
309	70	.10	2.10	2.00	885	880	880	882	313	307	307	309	211	1,188	806	.4639	.00341	67.40	85.41	15.6
310	55	.20	4.20	4.00	870	865	865	867	315	309	306	310	82	1,532	1,066	.7800	.00590	117.90	84.90	7.3
311	37	.30	6.30	6.00	890	875	880	882	299	293	294	295	131	1,112	965	1.0020	.00719	145.40	87.60	5.7
326	95	.00	.50	.50	910	870	875	885	234	232	230	232	77	1,808	555	.2070	.00133	27.05	97.05	27.51
318	70	.20	3.20	3.00	905	870	875	883	316	310	312	312	102	1,408	1,140	.6412	.00473	95.60	85.10	8.88
317	45	.30	5.20	4.90	905	870	880	885	291	289	289	290	81	1,190	938	.8400	.00595	120.50	88.30	6.78
312	45	.40	7.40	7.00	900	870	875	882	287	283	282	284	98	1,428	1,165	1.0260	.00722	146.00	89.40	5.55
313	45	.50	9.20	9.00	890	870	875	878	292	286	284	287	87	1,620	1,305	1.1620	.00834	166.10	89.20	4.90
314	45	.70	12.70	12.00	890	870	875	878	292	289	288	290	83	1,820	1,467	1.3100	.00938	189.20	88.40	4.35
315	50	.90	16.20	15.30	890	870	875	878	298	292	290	293	82	2,260	1,878	1.5100	.01087	219.10	88.00	3.77
316	45	1.00	19.10	18.10	880	860	875	872	300	293	290	294	84	2,200	1,777	1.5860	.01156	232.00	87.70	3.59

331	50	.80	19.90	19.10	920	900	905	905	908	912	304	299	297	300	304	80	214	120	2,406	2,200	1,7100	.01184	244.00	87.60	3.33
329	45	.80	22.10	21.30	920	900	905	905	912	912	307	302	300	304	304	80	214	81		2,066	1,8500	.01280	265.20	87.10	3.08
325	55	.00	.85	.85	1,245	1,170	1,185	1,185	1,200	1,200	271	269	270	268	268	76	211	(a)	748	717	.4620	.00209	52.20	94.25	12.32
327	60	.10	1.30	1.20	1,260	1,210	1,225	1,225	1,232	1,232	312	305	304	307	307	79	212	78	1,196	904	.6082	.00277	73.82	90.10	0.36
324	55	.10	2.20	2.10	1,245	1,190	1,185	1,185	1,207	1,207	355	352	352	353	353	79	211	86	1,442	1,133	.8260	.00407	102.30	85.72	6.89
328	45	.13	3.16	3.03	1,240	1,215	1,215	1,215	1,223	1,223	374	368	367	370	370	80	212	77	1,378	1,092	.9810	.00485	122.80	84.40	5.81
323	45	.20	4.14	3.94	1,225	1,180	1,180	1,180	1,195	1,195	354	352	353	353	353	80	212	95	1,426	1,250	1.1040	.00552	137.60	85.60	5.16
319	45	.40	5.70	5.30	1,260	1,200	1,215	1,215	1,225	1,225	327	324	323	325	325	71	213	108		1,568	1.3670	.00639	162.40	88.92	4.17
320	45	.50	8.50	7.80	1,230	1,180	1,185	1,185	1,198	1,198	324	320	319	321	321	74	213	89	2,080	1,735	1.5420	.00741	185.00	89.05	3.69
321	45	.70	12.70	12.00	1,230	1,185	1,195	1,195	1,203	1,203	331	326	327	328	328	78	215	85	2,486	2,133	1.9050	.00717	220.80	88.58	2.99
322	45	.90	18.00	17.10	1,240	1,200	1,210	1,210	1,217	1,217	340	335	334	336	336	80	217	103	2,926	2,672	2.3420	.01120	283.00	88.10	2.43
330	45	.90	21.40	20.50	1,240	1,210	1,210	1,210	1,220	1,220	345	338	336	340	340	81	218	106	3,176	2,883	2.5250	.01309	306.20	87.88	2.26
342	55	.00	.50	.50	1,565	1,460	1,470	1,470	1,498	1,498	246	244	245	245	245	80	211	(a)	1,000	625	.4025	.00135	39.90	97.35	14.15
341	45	.10	1.30	1.20	1,555	1,480	1,490	1,490	1,508	1,508	325	320	320	322	322	80	211	84	1,000	842	.7510	.00267	79.00	91.45	7.58
340	35	.20	2.50	2.30	1,555	1,510	1,510	1,510	1,525	1,525	405	396	399	400	400	82	212	92	1,304	1,018	1.1580	.00454	129.80	85.00	4.92
233	45	.10	2.70	2.60	1,580	1,520	1,525	1,525	1,542	1,542	394	387	386	389	389	76	211	90	1,600	1,200	1.0650	.00530	117.30	89.60	3.84
332	45	.25	4.20	3.95	1,510	1,485	1,485	1,485	1,492	1,492	409	402	402	404	404	78	212	103	1,930	1,614	1.4120	.00547	161.00	85.10	4.04
339	35	.30	5.30	5.00	1,515	1,475	1,465	1,465	1,485	1,485	396	388	391	392	392	82	213	107	1,666	1,274	1.4300	.00551	161.50	85.95	3.98
334	45	.80	6.20	5.90	1,555	1,525	1,520	1,520	1,533	1,533	396	387	390	391	391	80	214	128	2,340	2,020	1.7300	.00638	191.50	89.60	3.29
335	35	.80	9.60	9.10	1,535	1,505	1,500	1,500	1,513	1,513	372	365	365	367	367	80	216	107	2,112	1,978	2.2580	.00817	242.80	88.35	2.99
336	35	.60	12.60	12.00	1,550	1,520	1,510	1,510	1,527	1,527	375	368	370	371	371	82	217	106	2,368	2,245	2.5250	.00720	274.40	88.20	2.26
337	30	.80	17.00	16.20	1,515	1,490	1,480	1,480	1,495	1,495	375	368	365	369	369	83	219	106	2,278	2,204	2.8940	.01082	318.80	88.20	2.88
338	40	1.00	21.10	20.10	1,525	1,490	1,485	1,485	1,500	1,500	380	374	369	374	374	84	222	105	3,296	3,126	3.0820	.01153	340.50	88.15	1.85

a No water fed into the boiler during test.

TABLE 5.—Data and results of tests on boiler No. 4.

No. of test.	Duration of test (min-utes).	"Draft" (inches water).			Temperature (°F.).												Electrical energy consumed (watt-hours).	Water condensed (c. c.).	Heat absorbed per second (B. t. u.).	Air passed through boiler per second (pound).	Velocity of air entering boiler per second (feet).	True boiler efficiency (B _g).	Heating surface per boiler horsepower (square feet).
		Entering boiler.	Leaving boiler.	Difference.	Air entering boiler.			Air leaving boiler.						Air entering furnace.	Steam.	Feed water.							
					Thermometers—			Thermometers—			Average.												
					Average.			Average.															
					1	2	3	1	2	3													
348	30	0.30	0.80	0.50	713	693	678	695	397	386	389	391	66	211	...	0.728	393	0.464	0.00643	39.20	62.80	10.66	
349	45	.46	1.27	.81	704	685	672	687	409	399	404	404	65	211	73	1.380	596	.536	.00798	48.45	59.50	9.23	
350	45	.83	2.01	1.18	696	679	665	680	417	407	412	412	68	211	68	1.688	780	.706	.01110	66.60	57.12	7.01	
351	35	1.24	3.28	2.04	710	691	677	693	427	417	422	422	64	211	68	1.714	792	.922	.01432	87.00	56.50	5.37	
343	45	1.50	4.60	3.00	706	694	679	693	431	422	426	426	81	211	89	2.606	1,122	.997	.01572	95.50	55.34	4.96	
344	35	2.60	7.96	5.46	709	696	685	697	439	429	433	434	82	212	88	2.594	1,055	1.207	.01932	117.80	54.26	3.68	
346	35	3.90	10.90	7.00	713	694	673	693	434	423	427	428	65	212	94	3.376	1,210	1.372	.02181	132.50	55.05	3.61	
345	35	5.20	14.30	9.10	708	694	676	689	440	429	434	434	85	212	102	3.888	1,300	1.514	.02461	149.50	53.85	3.27	
347	35	6.42	18.33	11.91	712	688	666	689	435	423	427	428	66	213	85	3.888	1,417	1.622	.02618	158.40	54.90	3.05	
352	25	7.80	21.90	14.10	714	681	664	686	438	428	432	433	64	214	69	2.986	1,100	1.790	.02980	180.00	53.60	2.76	
359	30	.20	.70	.50	925	895	875	898	442	430	429	434	62	211	...	.876	505	.596	.00541	38.68	67.55	8.30	
360	35	.40	1.20	.80	905	875	860	880	454	449	447	450	63	211	87	.878	610	.698	.00684	48.30	64.32	7.08	
361	35	.60	1.90	1.30	925	885	860	890	478	469	466	471	65	211	77	1.574	758	.875	.00880	62.52	61.68	5.66	
353	35	1.20	3.35	2.15	925	895	875	898	509	481	478	489	63	212	64	2.154	955	1.116	.01149	82.15	59.68	4.03	
354	35	1.75	4.98	3.23	925	905	875	902	523	495	478	499	64	212	66	2.612	1,190	1.389	.01451	104.20	58.44	3.56	
355	45	2.10	5.90	3.80	940	920	880	913	508	498	494	500	57	211	112	3.690	1,815	1.577	.01489	107.70	58.80	3.18	
356	35	3.20	9.20	6.00	940	915	880	912	518	507	503	509	61	214	87	3.540	1,595	1.826	.01908	137.80	57.75	2.71	
357	40	5.10	14.50	9.40	950	905	870	908	512	499	497	503	61	216	82	4.660	2,038	2.050	.02160	153.60	58.50	2.41	
358	30	7.10	20.30	13.20	910	885	850	882	504	489	487	493	60	216	124	4.130	1,862	2.403	.02604	183.70	58.42	2.06	
362	35	.20	.70	.50	1,210		1,185	1,198	538	525	524	529	67	211	75	1.280	827	.956	.00602	40.80	67.78	5.18	
363	35	.50	1.30	.80	1,230	1,200	1,190	1,207	584	578	575	579	68	212	82	1.748	1,008	1.157	.00776	53.00	63.10	4.28	
364	30	.60	1.80	1.20	1,250	1,195	1,195	1,180	608	604	595	602	67	213	94	1.838	1,040	1.378	.01004	86.70	59.85	3.59	

INTERRELATION OF DATA AND RESULTS.

Figures 4 to 63 have been platted to show graphically the relations between the various items of the observed data of the experiments, and also the relations of the observed data to the calculated results. The fact that in some of the figures the points do not lie close to the smooth curve passed through them may be accounted for by errors in temperature measurements, and by slight variations of the initial temperature from the constant figure intended.

The worst errors in temperature readings may amount to 5 per cent of the temperature recorded. These errors were not due to incorrect instruments, but rather to the location of the thermometers or the junctions of the thermocouples, and to the influence of radiation on the instruments. Although the three thermocouples and thermometers used for measurements of the initial and final temperatures were within 1 inch of one another and were screened from the boiler and the furnace, their readings differed at times as much as 50° F. at high initial temperatures. This discrepancy in temperature readings varied with the velocity of the air flowing through the apparatus, becoming less as the velocity increased.

The average initial temperatures of individual tests in each series varied about 3 to 6 per cent on either side of their intended constant values. The variations crept in notwithstanding the precautions taken to get the temperatures constant. The cause of this trouble seems to have been the high heat capacity of the furnace, although the latter was built as compactly as possible. When, in the case of the higher temperatures, the current in the furnace was reduced, the temperature stayed up for some time after the reduction of the current, and sometimes even continued to rise for a minute or two.

RELATION OF THE INITIAL VELOCITY OF THE AIR TO THE RATE OF HEAT ABSORPTION AND TO THE TRUE BOILER EFFICIENCY.

In figures 4 to 7 the upper sets of curves show the relation of the initial velocity of the air to the rate of heat absorption, and the lower sets of curves the relation of the initial velocity to the true boiler efficiency. Figure 4 shows these relations for boiler No. 1, figure 5 for boiler No. 2, figure 6 for boiler No. 3, and figure 7 for boiler No. 4. Each curve on the charts is drawn through a group of points representing tests at a constant initial temperature, as indicated by the figure near each curve. The initial velocity is the calculated velocity of the air as it enters the flues of the boiler.

The upper sets of curves of all four figures indicate that the quantity of heat absorbed by each boiler per second increases almost directly with the initial velocity of the air. In each figure the tests

having nearly the same initial temperature fall along the same smooth curve, which is almost a straight line.

The lower sets of curves of all four figures indicate, in general, that, as the initial velocity of air increases, the true boiler efficiency drops, at first very rapidly; and then, after a certain velocity is reached, it becomes nearly constant for tests with low initial temperatures, and varies but little for those with higher initial temperatures.

As an illustration of a rather small variation in true boiler efficiency for a considerable range in capacity, the results of two sets of tests made on boiler No. 2 are given (fig. 5). Taking the initial temperature of 1,200° F. and the initial velocity of 85 feet per second, the

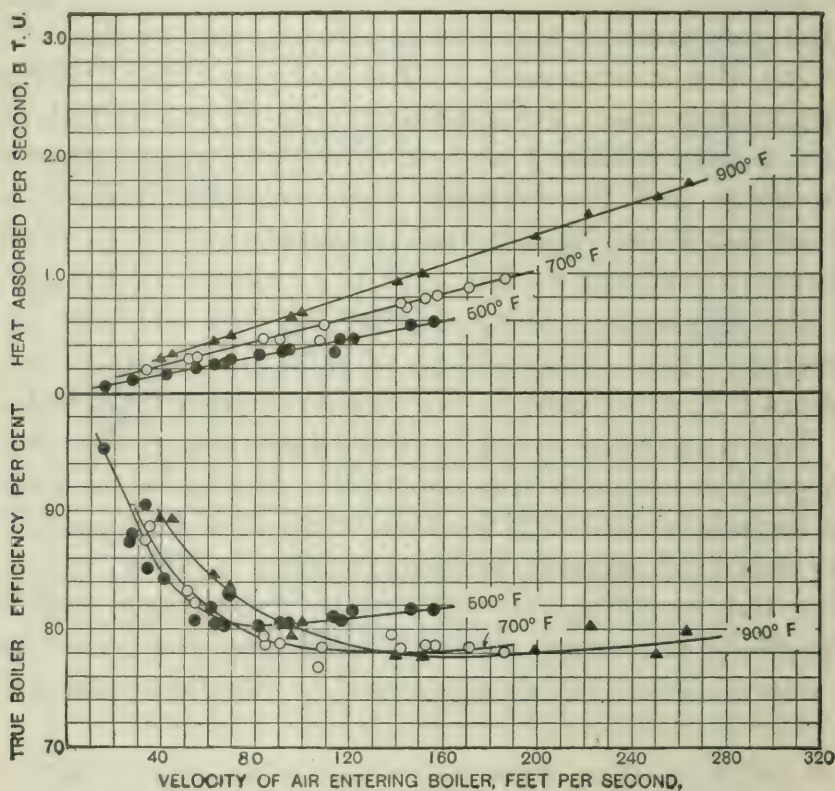


FIGURE 4.—Relation of velocity of air entering boiler to rate of heat absorption and the true boiler efficiency. Boiler No. 1.

true boiler efficiency is 70 per cent, and the rate of heat absorption about 0.96 B. t. u. per second. Then, taking the same initial temperature, but an initial velocity of 335 feet per second, the true boiler efficiency is about 61.2 per cent, and the rate of heat absorption 3.3 B. t. u. per second. In the latter case the capacity has been increased 3.48 times, whereas the efficiency has dropped only 8.8 per cent, or 12.6 per cent of itself.

For any boiler and any initial temperature the velocity at which the true boiler efficiency curve becomes nearly horizontal is perhaps approximately the velocity at which eddyding of the gas appears; this is termed by physicists the critical velocity. Up to this critical velocity the stream lines of the air flowing through the tubes remain

parallel to each other, that is, there is no other mixing of the air than the natural diffusion. Under these conditions the air near the metal, already cooled, tends to stay there, and the air near the center of the tube, still hot, continues to flow through the center without having a chance to come near and give up heat to the metal of the tube. However, as soon as the critical velocity of the air is reached the parallel streams break up and there is a constant interchange of the particles of gas next to the metal and those near the

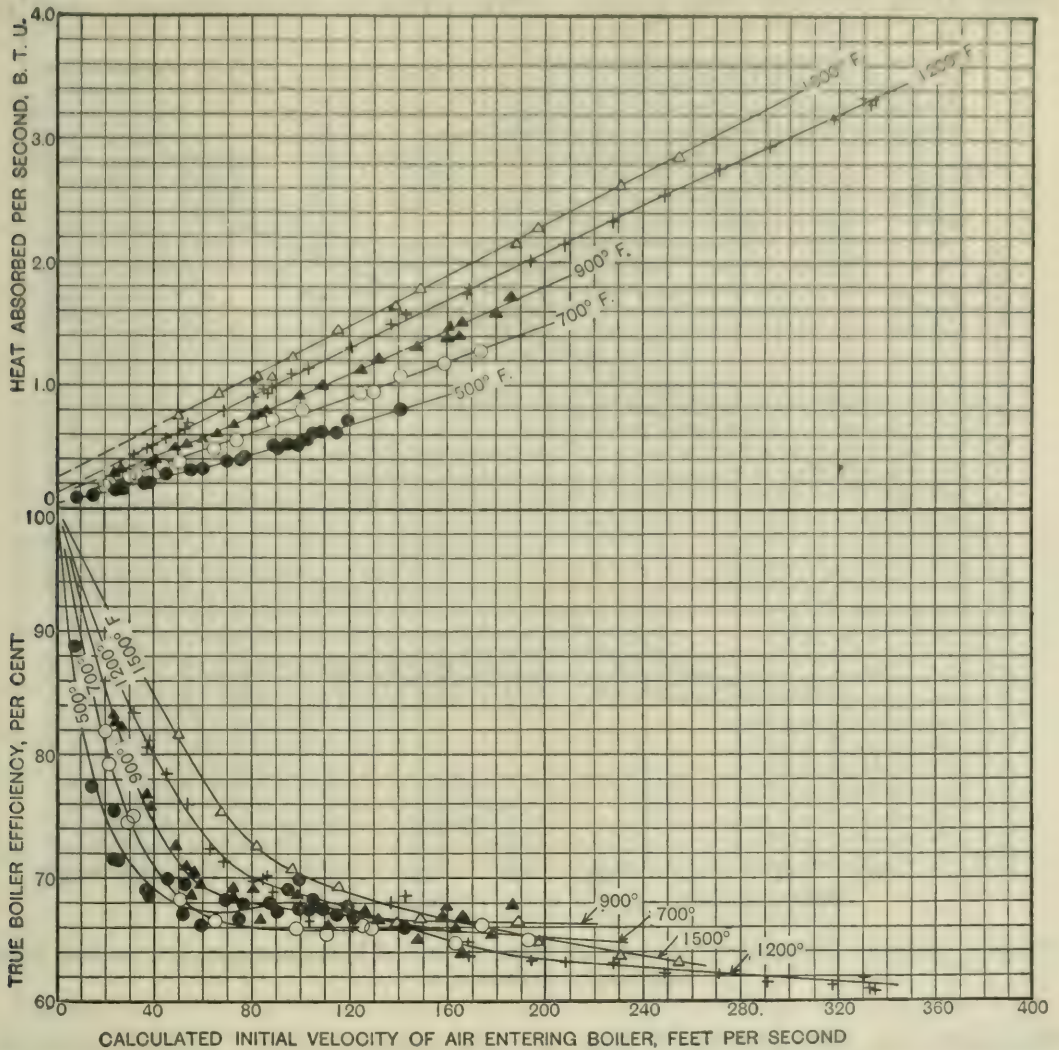


FIGURE 5.—Relation of velocity of air entering boiler to rate of heat absorption and the true boiler efficiency. Boiler No. 2.

center of the flues, so that the molecules alternately approach the metal and give it their heat energy.

As shown by the curves, the critical velocity varies with the initial temperature of the air; the higher the initial temperature, the higher the critical velocity. With high initial temperature the critical velocity is not so well defined as it is with initial temperatures of 500° and 700° F. For very high initial temperatures, such as are sometimes used in boiler practice (2,500° to 3,000° F.) the bend in the curve probably extends over a considerable range of veloc-

ity; hence it is possible that in such cases the boiler never works on the horizontal portion of the curve.

By comparing the lower sets of curves of figures 4 to 7, it is apparent that the critical velocity varies also with the diameter of the flues but apparently not with their length.

Although with low initial temperatures the parts of the true boiler efficiency curves beyond the critical-velocity point is almost perfectly horizontal, with high initial temperatures the drop seems to be constant even beyond this point. This drop in the efficiency can be

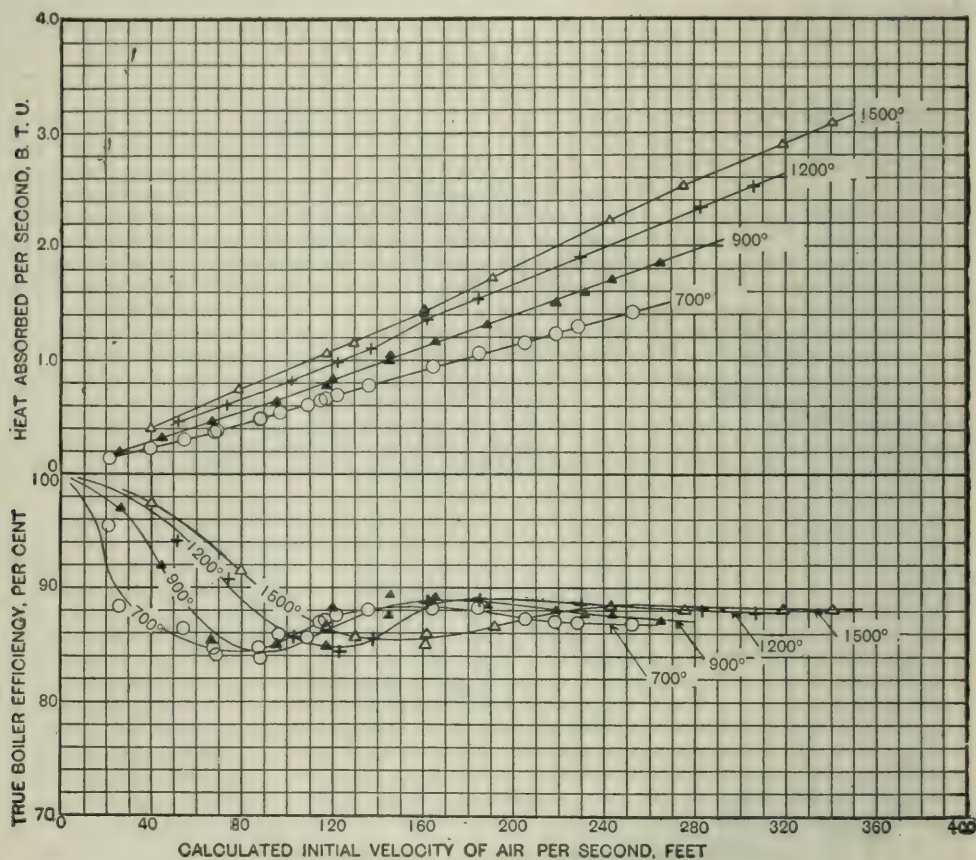


FIGURE 6.—Relation of velocity of air entering boiler to rate of heat absorption and the true boiler efficiency. Boiler No. 3.

explained by the increased rate of heat absorption. With the same conditions of the heating plate the quantity of heat transmitted through the plate in a unit of time depends upon the temperature difference between the dry and the wet surface. The only way in which this temperature difference can be increased is by raising the temperature of the dry surface. The dry surface, however, cools the moving gases; therefore, if the temperature of the dry surface is raised the gases leave the boiler at a relatively higher temperature than if the rate of heat absorption is lower; hence the true boiler efficiency is lower.

The lower set of curves of figure 6 shows one very peculiar feature. As the initial velocity of the air increases, the true boiler efficiency at first drops rather rapidly, then rises 2 or 3 per cent, and then remains nearly constant for a considerable range of initial velocity. As this depression in the curves was difficult to explain, the thermometers were taken out of the apparatus, very carefully put in again to the right depth, and check tests run. The results of these

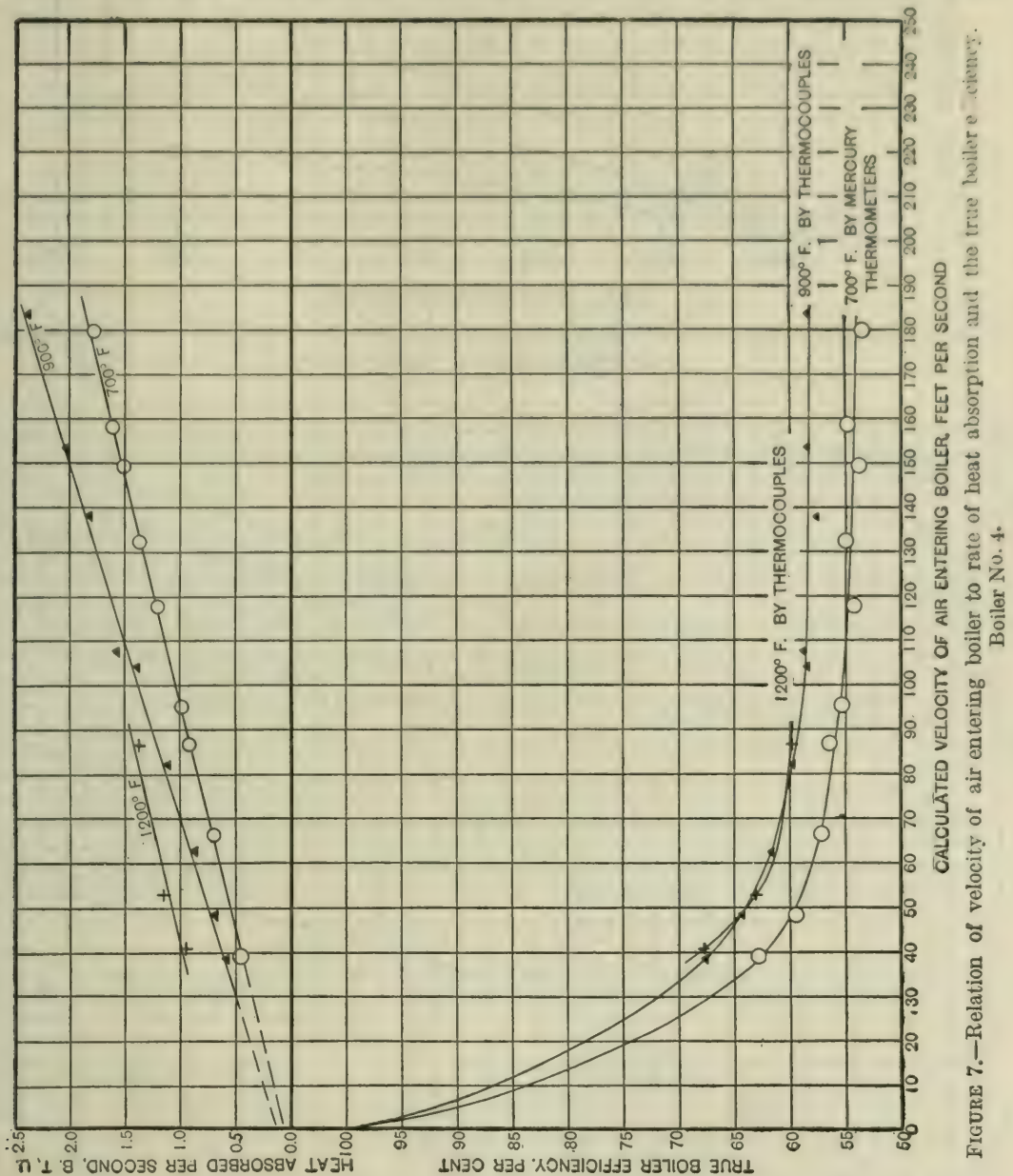


FIGURE 7.—Relation of velocity of air entering boiler to rate of heat absorption and the true boiler efficiency.

Boiler No. 4.

check tests, however, came very close to the results of the tests run before.

There is one indication in the efficiency curves that deserves particular attention; this is that for any one boiler the true boiler efficiency, after the gases have passed the critical velocity, is very nearly the same for all initial temperatures. It may be stated, then, that *the true boiler efficiency of any one boiler is practically independent of the initial temperature of the gases.*

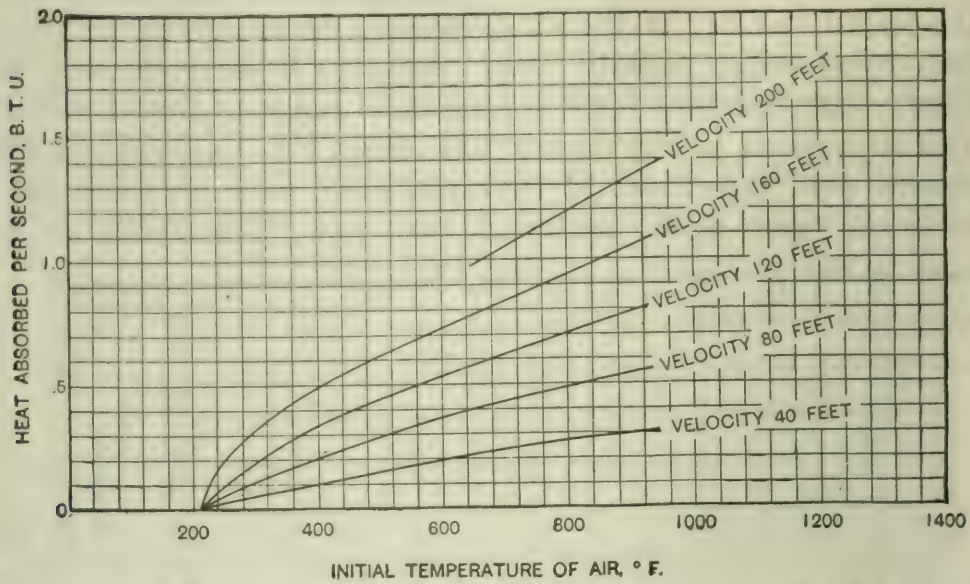


FIGURE 8.—Relation of temperature of air entering boiler to rate of heat absorption when initial velocity of air remains constant. Boiler No. 1.

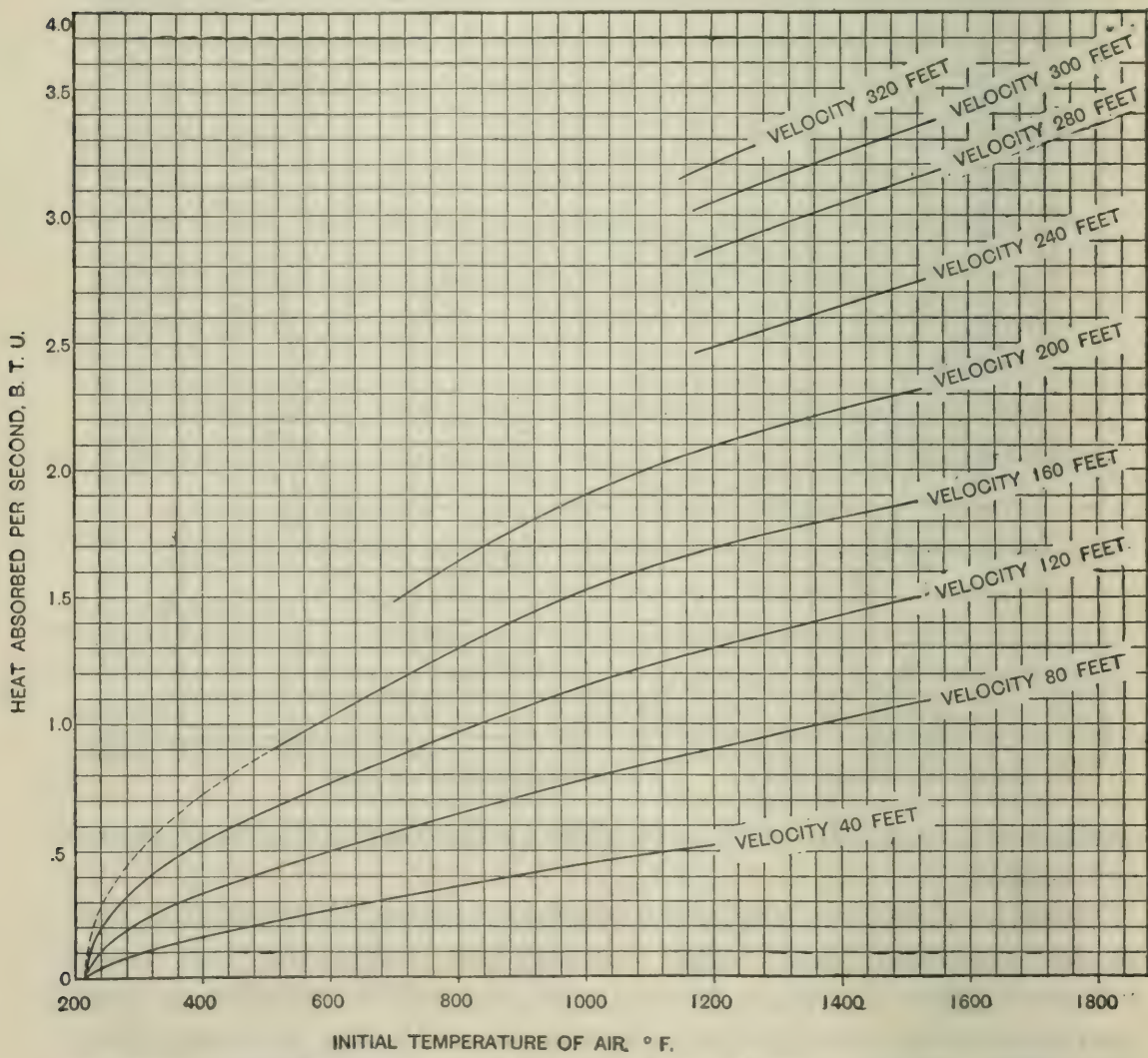


FIGURE 9.—Relation of temperature of air entering boiler to rate of heat absorption when initial velocity of air remains constant. Boiler No. 2.

RELATION BETWEEN INITIAL TEMPERATURE AND RATE OF HEAT ABSORPTION.

Figures 8 to 11 show the effect of variations in the initial temperature of air on the rate of heat absorption when the initial velocity of the air remains constant. Figure 8 shows this effect for boiler No. 1, figure 9 for boiler No. 2, figure 10 for boiler No. 3, and figure 11 for boiler No. 4. Each curve gives this effect for one constant velocity, as indicated by the figures. The points for the curves were obtained by interpolation from the upper curves of figures 4 to 7.

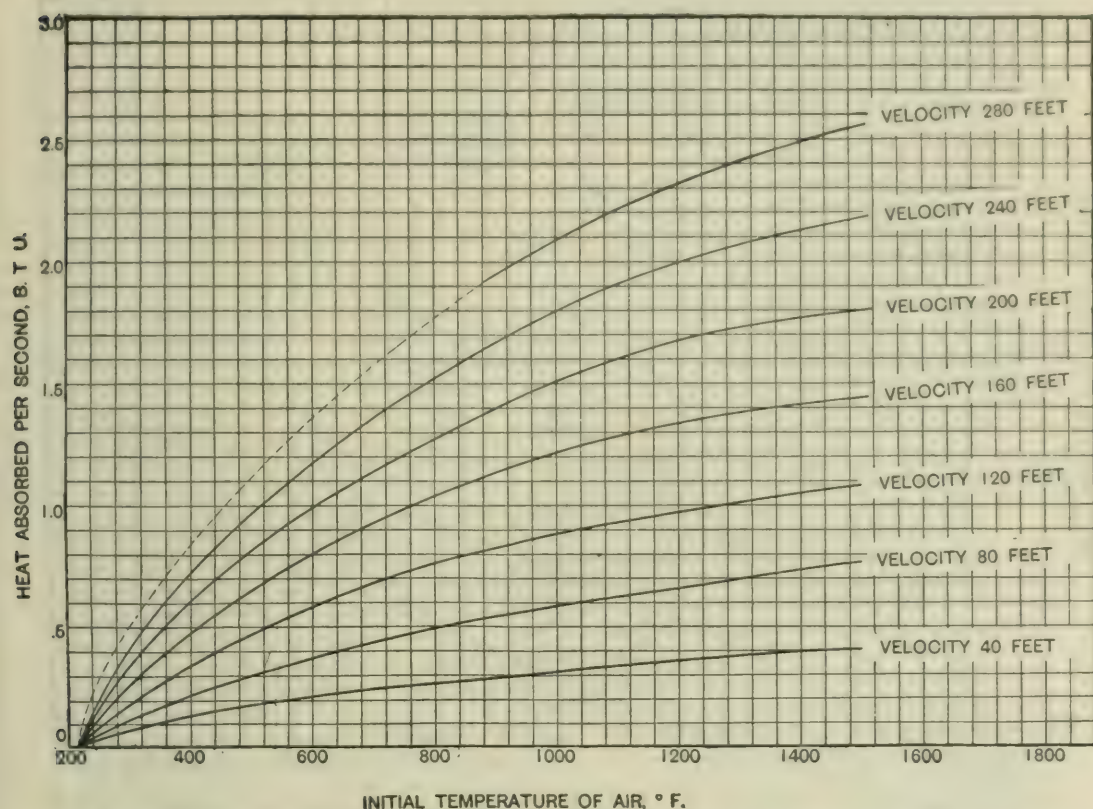


FIGURE 10.—Relation of temperature of air entering boiler to rate of heat absorption when initial velocity of air remains constant. Boiler No. 3.

The curves of figures 8 to 11 have, roughly, a parabolic shape. They indicate that as the initial temperature of the air rises the rate of heat absorption increases, but not in proportion to the temperature. At lower temperatures there is a considerable increase in the quantity of heat absorbed for each increment in temperature, but as the temperature becomes higher the corresponding increase of heat absorbed for equal increments in temperature becomes less, until probably at very high temperatures there would be little or no gain in the rate of heat absorption from a further rise of the air temperature. This fact indicates that there is another factor influencing the rate of heat absorption—a factor which varies inversely

as the temperature. This factor, the density of the gas, will be discussed more fully in the general conclusion from these experiments (p. 114).

RELATION OF THE WEIGHT OF AIR PASSING THROUGH THE BOILER TO THE RATE OF HEAT ABSORPTION AND TO THE TEMPERATURE OF THE AIR LEAVING THE BOILER.

The upper sets of curves of figures 12 to 15 give the relation between the weight of air passing through the boiler and the rate of heat absorption; the lower sets give the relation between this weight

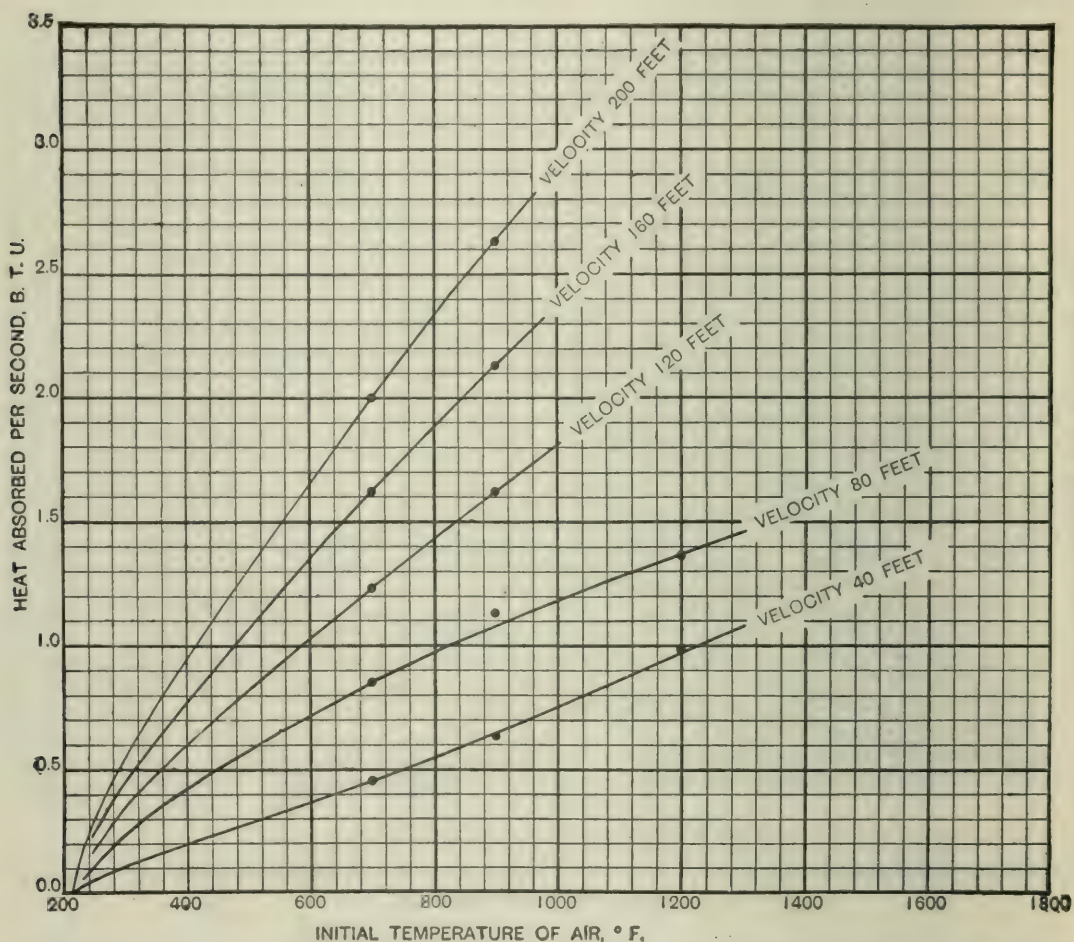


FIGURE 11.—Relation of temperature of air entering boiler to rate of heat absorption when initial velocity of air remains constant. Boiler No. 4.

of the air and the temperature of the air as it leaves the boilers. Each curve is drawn through points that represent tests in which the air was kept at a constant initial temperature, as indicated by the figures near the curve. The number of the boiler on which the tests were run is given in the legend for each figure. The weight of air passing through the boiler was calculated by the method given on page 31.

The upper sets of all four figures show that the quantity of heat absorbed by any one of the four boilers increases almost in direct proportion to the weight of air passing through the boiler. This is

an important relation that can be used to advantage in steam-boiler practice.

A boiler can be made to respond to an increased demand for steam by increasing the weight of hot gas passing through it. More hot gas can be produced if the rate of combustion is increased. Of course, to put a larger weight of gas through a boiler, the difference between the gas pressures at the two ends of the boiler must be increased.

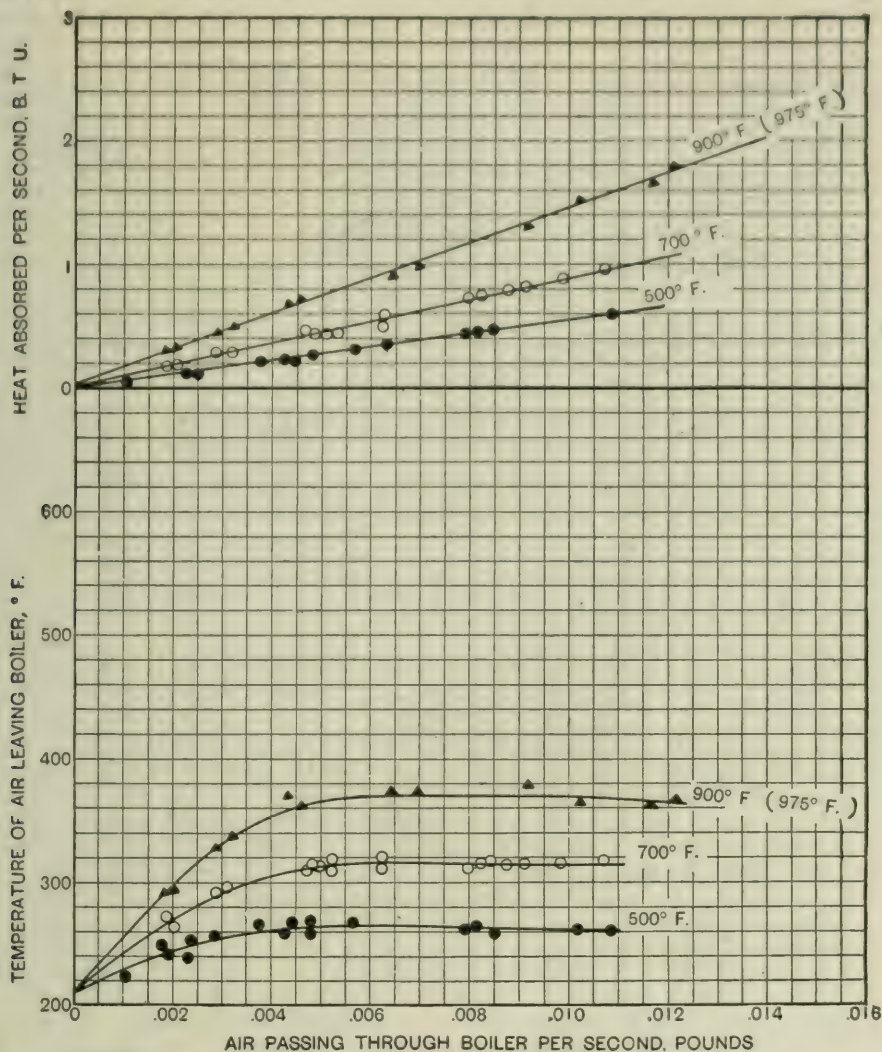


FIGURE 12.—Relation of the weight of air passing through boiler to the rate of heat absorption and the temperature of air leaving boiler. Boiler No. 1.

The lower sets of curves of figures 12 to 15 show, for every group of tests in which initial temperature of the air was constant, that as the weight of air passing through the boiler increases the temperature of the air leaving the boiler rises, at first very rapidly, then less and less so, until after a certain value of the weight of air is reached the temperature remains nearly constant. Especially is this true for tests in which the initial temperature of the air was low.

The curves of final temperatures for boiler No. 3 (fig. 14) make apparent in another way the same peculiarity of this boiler that was shown in figure 6 by the true boiler efficiency curves, this peculiarity

being that after the leaving temperatures had risen to a maximum they dropped 20° or 30°, and then became nearly constant.

As stated before, check tests persistently showed the hump in the curves for those particular values of the weight of air. No explanation of this feature can be offered at present.

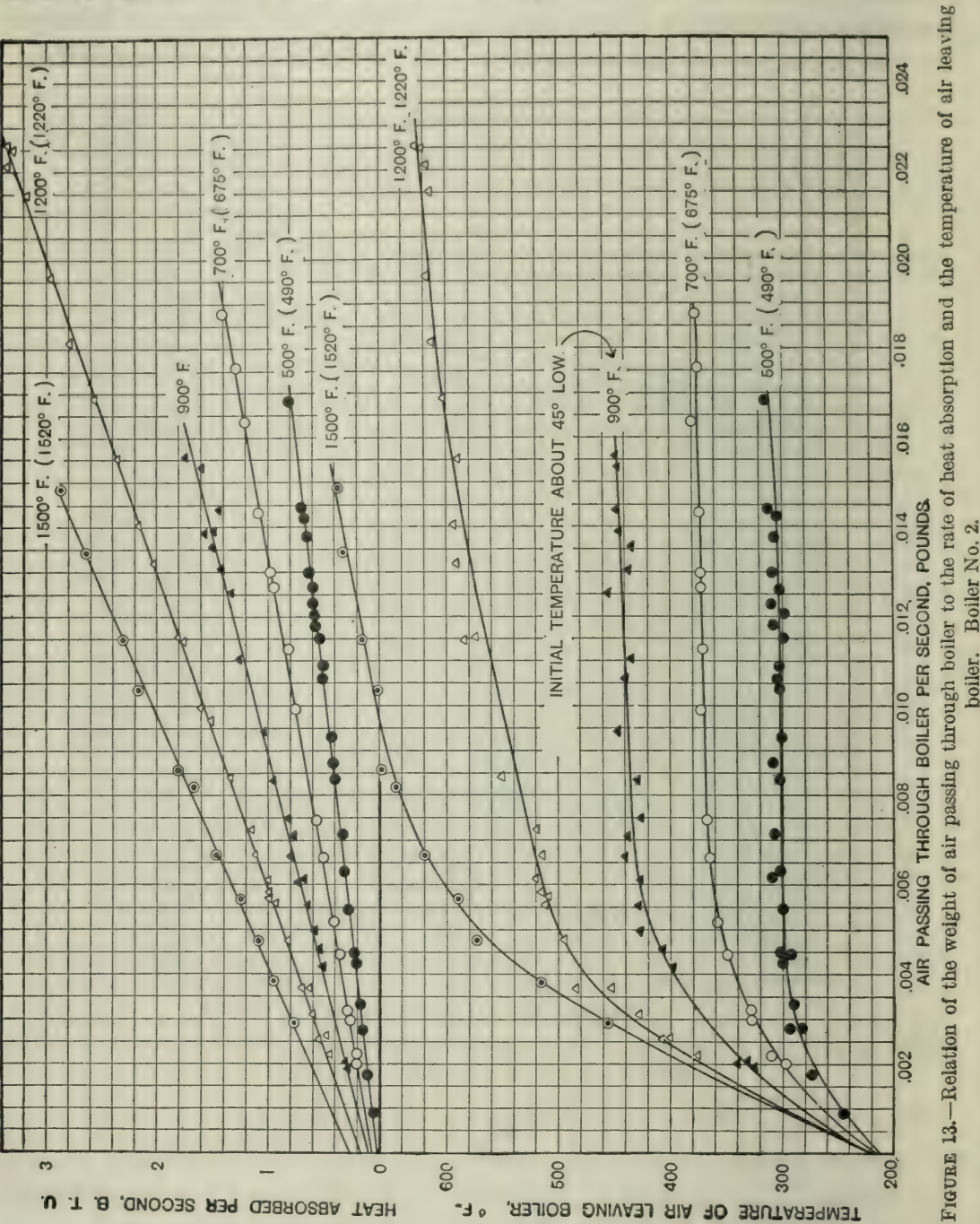


FIGURE 13.—Relation of the weight of air passing through boiler to the rate of heat absorption and the temperature of air leaving boiler, Boiler No. 2.

RELATION OF TRUE BOILER EFFICIENCY TO THE DIMENSIONS OF FLUES.

Comparison of the true boiler efficiency of the tests made with the first four boilers brings out the fact that the tests with boiler No. 3 have a higher true boiler efficiency than those with boiler No. 1, that the tests with boiler No. 1 have a higher true boiler efficiency than those with boiler No. 2, and that those made with boiler No. 2 have

a higher true boiler efficiency than those made with boiler No. 4. Thus the average of the true boiler efficiencies for boiler No. 3 is about 88 per cent, for boiler No. 1 about 78 per cent, for boiler No. 2 about 66 per cent, and that for boiler No. 4 about 56 per cent.

Boilers Nos. 2 and 4 were of the same length as boiler No. 1, but their flues were larger in diameter than those of boiler No. 1. It may be said, then, that if the length of the flues is constant, the true boiler efficiency drops when the diameter of the flues is increased.

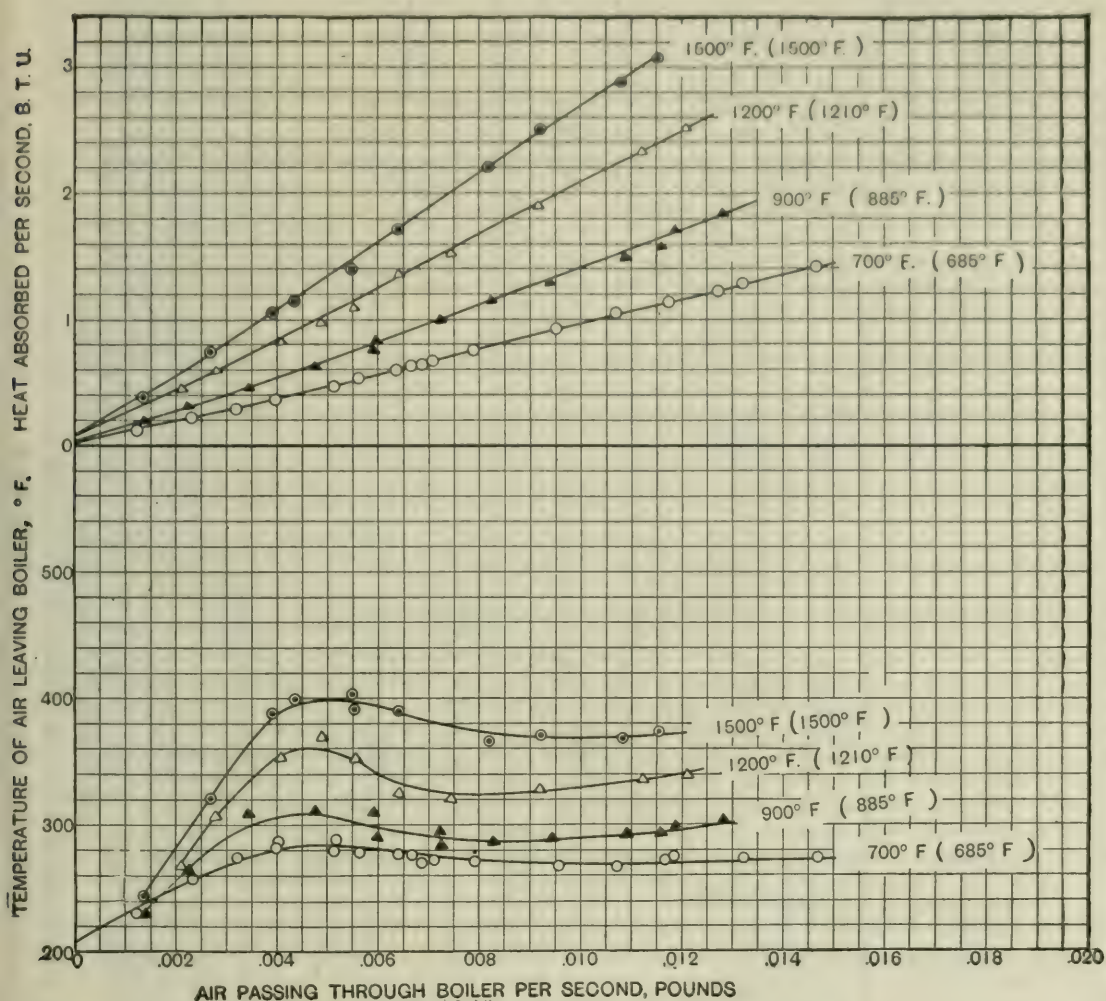


FIGURE 14.—Relation of the weight of air passing through boiler to the rate of heat absorption and the temperature of air leaving boiler. Boiler No. 3.

As only three different diameters of flues were tested, no exact relation can be deduced from the results of the experiments.

The diameter of the flues of boiler No. 3 was the same as that of the flues of boiler No. 1, but the former boiler was twice as long as the latter. It may be said, therefore, that if the diameter of the flues remains constant the true boiler efficiency increases with the length of the flues, although not in direct ratio. Each successive equal increase in the length of flues increases the true boiler efficiency less than the previous increase.

Figure 16 shows further the influence of the length and the diameter of flues on the true boiler efficiency. The chart gives the final

temperature for the four boilers when the initial temperature was 900°F . and the same weight of air (about 0.01 pound) passed through each boiler per second. The curves indicate the drop in temperature of the air as it passed through the flues. The platted values of the

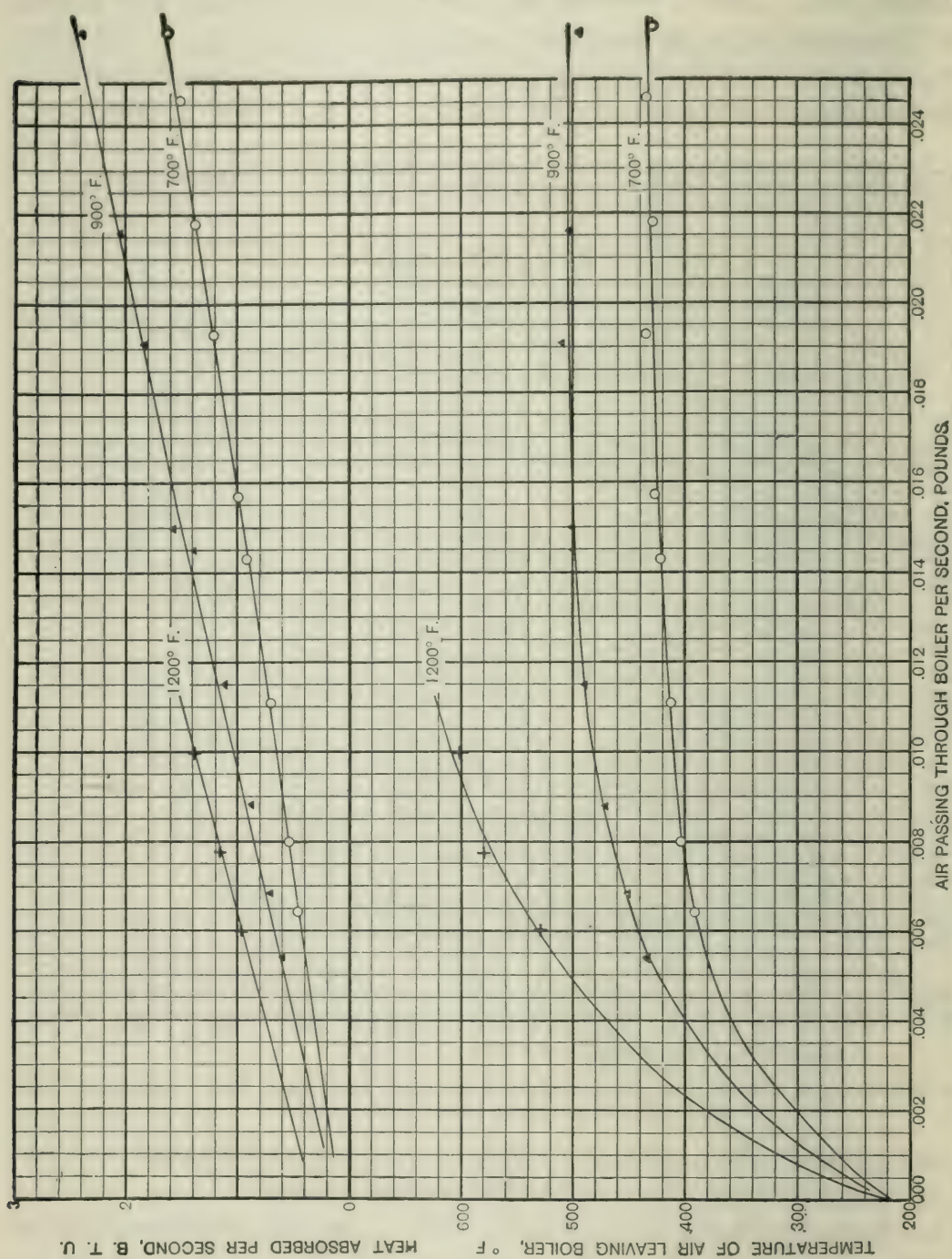


FIGURE 15.—Relation of the weight of air passing through boiler to the rate of heat absorption and the temperature of air leaving boiler.
Boiler No. 4.

final temperature of the air have been taken from the data obtained in the experiments. The curves connecting these points and representing the temperature drops are only approximate, although they are perhaps close to their actual shape. The upper curve represents the temperature drop through boiler No. 4 and the middle curve

through boiler No. 2. The lower curve from 0 to 8 inches of the boiler flues represents the temperature drop through boiler No. 1, and from 0 to 16 inches the drop through boiler No. 3. All curves are drawn asymptotic to the horizontal line representing steam temperature; that is to say, if the flues (boilers) of either diameter were made infinitely long, the temperature of the air passing through them would be reduced to that of the steam. The chart shows that although the large flues of boiler No. 2 have more heating surface than the small flues of boiler No. 1, the air comes cooler from the smaller flues than from the larger ones. This is an important feature which brings

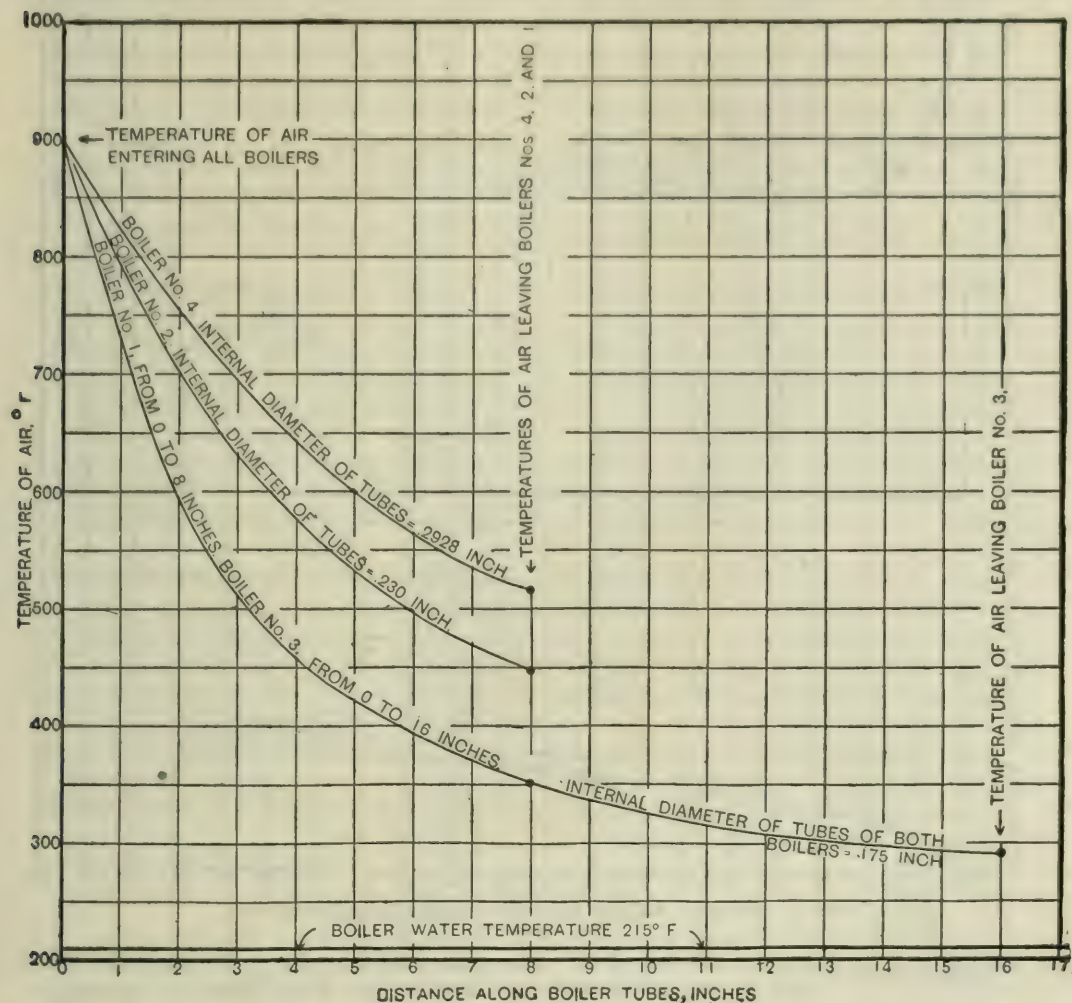


FIGURE 16.—Effect of the length and the diameter of the flues on the temperature of the air leaving boiler.

out the fact that *it is not area but arrangement that makes heating surface efficient.*

The same chart shows that increasing length of flues beyond a reasonable limit does not pay. The double-length flues of boiler No. 3 lowered the final temperature of the air only about 70° F. more than did the single-length flues of boiler No. 1; or, boiler No. 3 absorbed only about 8 per cent more heat than boiler No. 1, notwithstanding the fact that the former had twice as much heating surface as the latter.

RELATION OF THE EFFECTIVE PRESSURE DROP TO THE TRUE BOILER EFFICIENCY AND THE WEIGHT OF GASES PASSING THROUGH THE BOILERS.

In figures 17 to 20 the lower curves show the relation between the pressure drop of the air in passing through the boiler and the weight of the air passing through. The upper sets of curves give the relation between the pressure drop and the true boiler efficiency. The

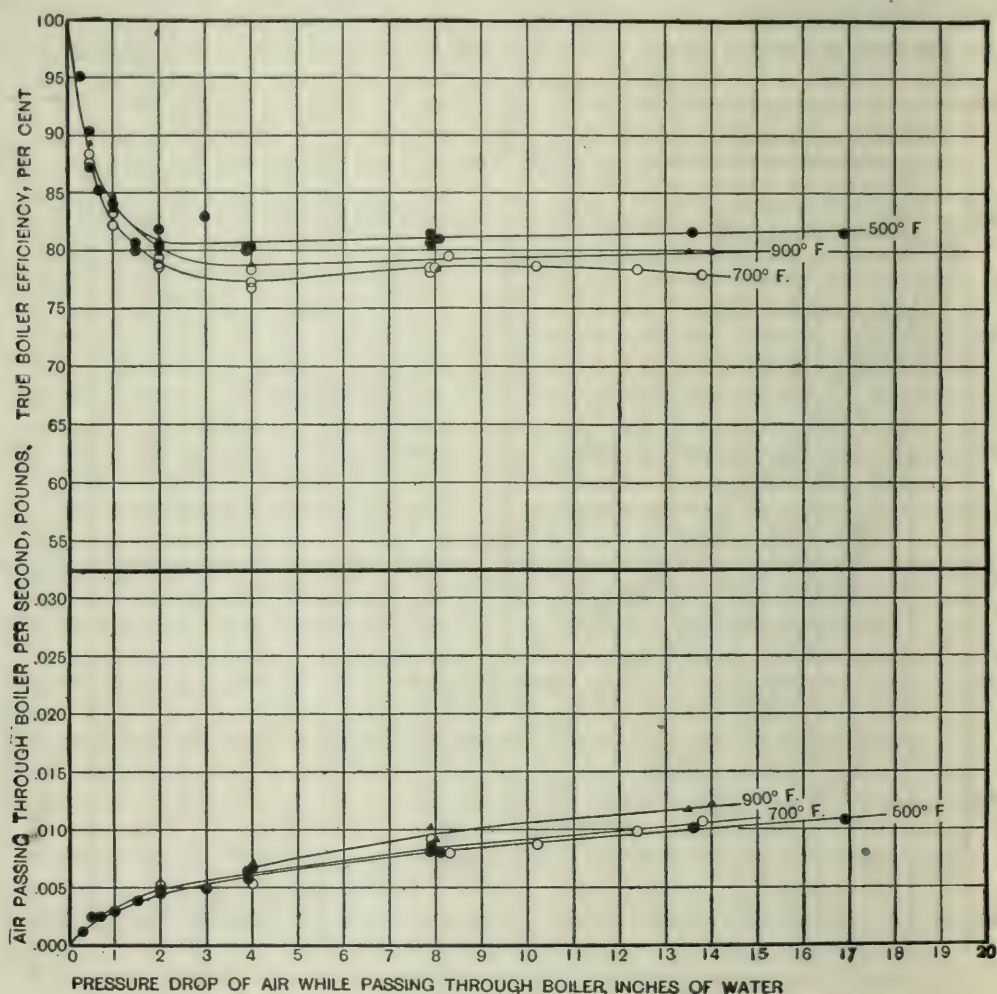


FIGURE 17.—Relation of the pressure drop of air while passing through boiler to the true boiler efficiency and the weight of air passing through boiler. Boiler No. 1.

weight of air was calculated according to the method outlined on page 31. The pressure drop through the boiler is the difference between the pressure of the air entering the boiler and that of the air leaving the boiler, and is commonly spoken of as the “draft” lost in the boiler.

The lower sets of curves were platted as a check on the calculated velocity and weight of air used in other charts of this discussion. With a constant initial temperature, the weight of air passing through the boiler must have some constant relation to the difference of pres-

sure at the two ends of the flues. The curves indicate that such a relation does exist; they have a shape similar to parabolas of the equation $y^2=kx$. With a constant initial temperature, the initial velocity varies directly as the weight of air passing through the flues, and therefore the same relation exists between the initial velocity

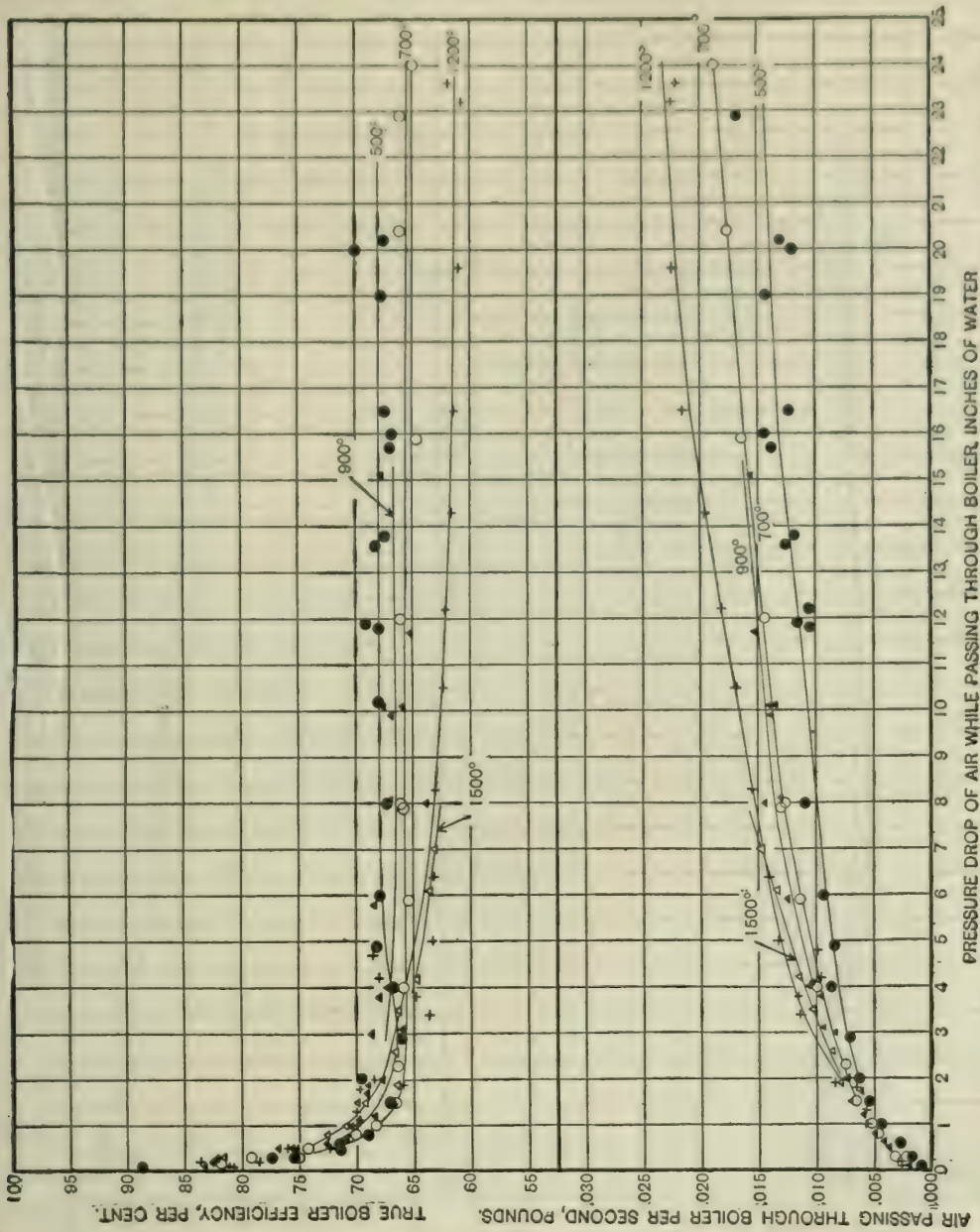


FIGURE 18.—Relation of the pressure drop of air while passing through boiler to the true boiler efficiency and the weight of air passing through boiler. Boiler No. 2.

and the pressure drop. According to the curves of figures 17 and 18, the weight of air passing through the flues of the boilers increases also with the initial temperature, although not in direct proportion. The air-weight curves of figures 19 and 20, on the contrary, show decreases of the weight of air for higher initial temperatures. This last feature is probably due to the fact that boiler No. 3 was twice as long as boilers Nos. 1 and 2; and in the case of boiler No. 4 to the

fact that the sets of temperatures of 700° and 900° F. were not really 200° F. apart.

The upper sets of curves of figures 17 to 20 show the effect of pressure drop on the true boiler efficiency. In shape the curves are somewhat similar to the lower sets of curves of figures 4 to 7. The true boiler efficiency decreases at first very rapidly when the pressure

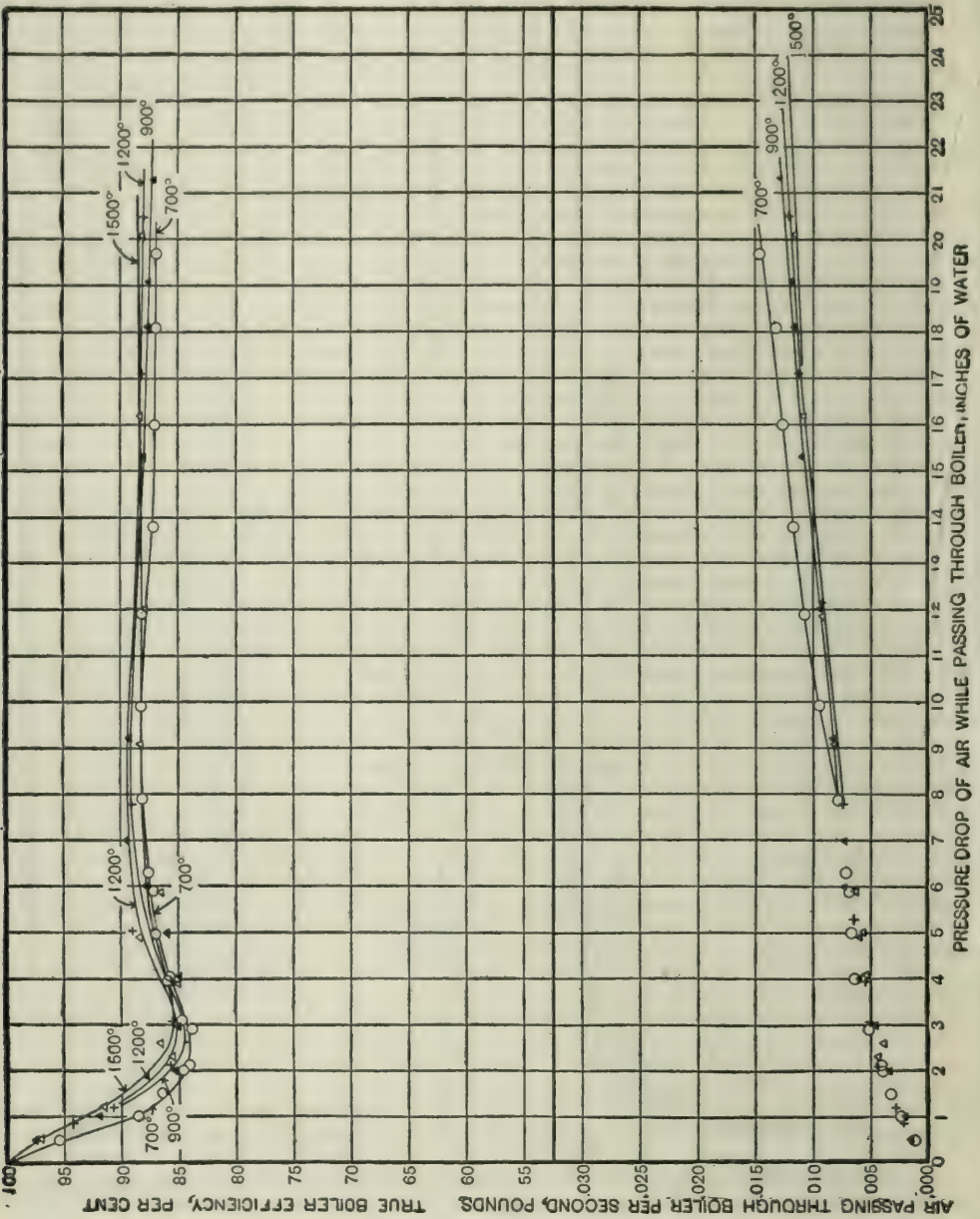


FIGURE 19.—Relation of the pressure drop of air while passing through boiler to the true boiler efficiency and the weight of air passing through boiler. Boiler No. 3.

drop increases, but after the latter passes a certain value, which is different for each boiler, the efficiency remains nearly constant for the full range of the experiments. The slight continual drop beyond the bends in the efficiency curves of the higher initial temperatures may be accounted for by the rapidly increasing capacity, as explained in connection with figures 4 to 7.

It may be questioned why at the left-hand side of the figures the curves were prolonged to a point of 100 per cent true boiler efficiency when the pressure drop for the velocity is zero. This was done because with no velocity the air would remain in the boiler an infinite

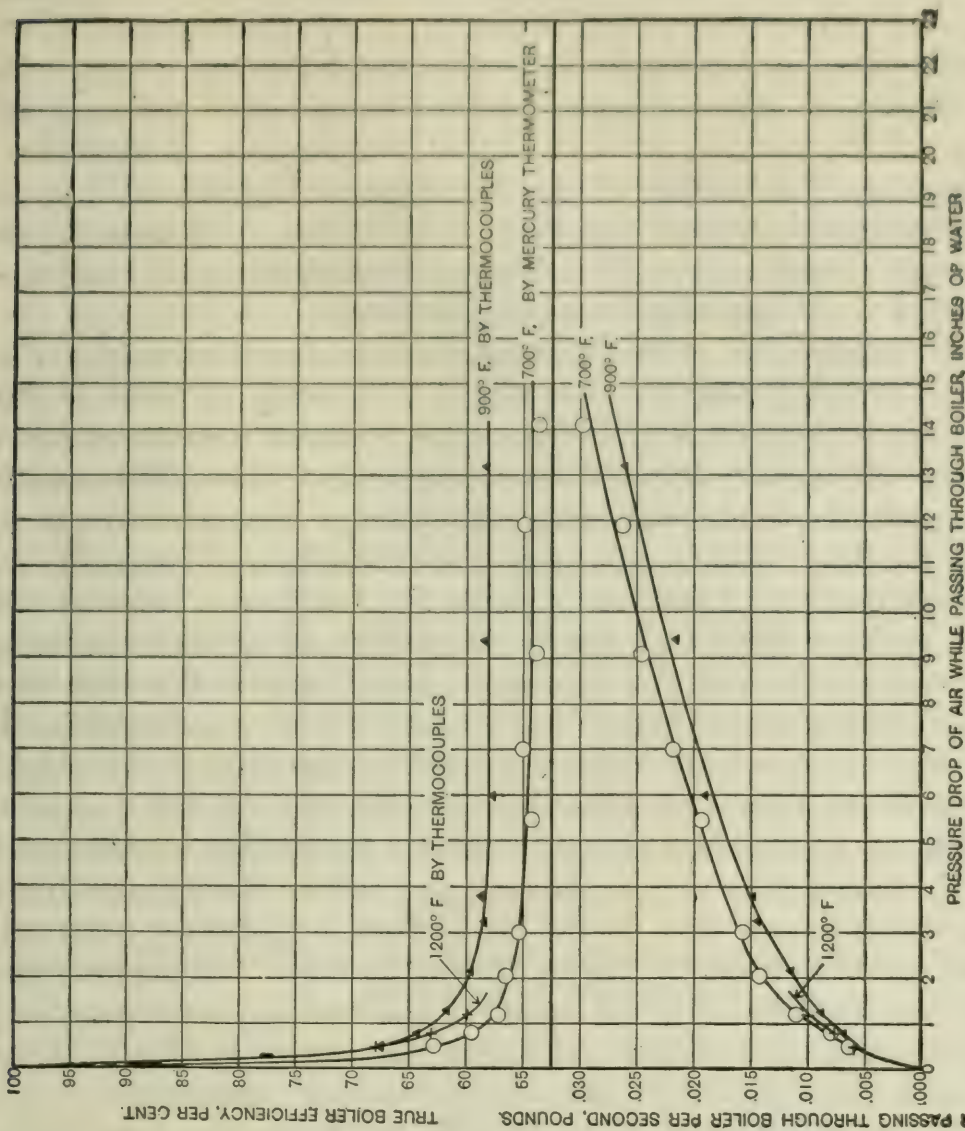


FIGURE 20.—Relation of the pressure drop of air while passing through boiler to the true boiler efficiency and the weight of air passing through boiler. Boiler No. 4.

length of time, and its temperature would finally be reduced to that of the steam in the boiler, making the true boiler efficiency equal to 100 per cent.

The curves of the lower set of figure 19 show the same peculiar hump as the efficiency curves of figure 6; as stated before, no explanation can be offered at present.

RELATION OF THE DIAMETER OF FLUES TO WEIGHT OF AIR PASSING THROUGH THEM WHEN THE LENGTH OF THE FLUES IS CONSTANT.

Comparison of the lower sets of curves of figures 17 and 18 shows that on account of the flues being larger a smaller pressure drop through boiler No. 2 is required than through boiler No. 1 to push the same weight of air at the same initial temperature through both boilers. For example, with an initial temperature of 900° F., a pressure drop of 4.5 inches of water is required to push 0.075 pound of air per second through the flues of boiler No. 1, whereas a pressure drop of only about 2 inches is required to push the same amount of air through the flues of boiler No. 2.

Table 6 shows that with the same pressure drop the weight of air flowing through boiler No. 2 is about 1.48 times as great as that flowing through boiler No. 1. This ratio is almost a constant within the range of pressure drops of the experiments. It would seem that this ratio should be the same as, or rather larger than, the ratio of the area of the flues; that is, since the area of the flues of boiler No. 2 is 1.72 times the area of the flues of boiler No. 1, the same pressure drop should perhaps push 1.72 times the weight of air through boiler No. 2 as through boiler No. 1, or even more, because the velocity of approach is higher for boiler No. 2 than for No. 1. However, the ratios are not the same. Table 6 shows that the ratio of the weights of air is about 14 per cent smaller than the ratio of the flue areas. One rational explanation of the ratios not being alike in this case is that the smaller flues absorb the heat and reduce the temperature of the air much sooner than do the larger flues, and thereby reduce the volume and the velocity of the air flowing through. As the resistance due to friction varies with the square of the velocity, a comparatively small reduction of the velocity of the air through the smaller flues may result in a considerable decrease in the frictional resistance.

Table 7 shows the same characteristic for boiler No. 4 as Table 6 does for boiler No. 2—the weight of air flowing through the larger tubes of boiler No. 4 is less than might be expected from their greater area. It is not at all certain that the initial temperatures of 900° F. in the tests on this largest-tubed boiler were close to 900° F.; but, on the assumption that the temperatures plotted are correct, the quantities of air that flowed through this boiler under the several pressure drops averaged 2.22 times the quantities that flowed through boiler No. 1. The area of the larger tubes was 2.705 times that of the smaller, hence the quantities of air that passed through the former fell 18 per cent short of what might have been anticipated.

Inasmuch as the tests with boiler No. 4 at 700° F. are more trustworthy than those at 900° F., Table 8 is presented for boilers 1 and

4 at 700° F. initial air temperature. In this case it turns out that the weights of air flowing through the larger tubes average 2.77 times the weights flowing through the smaller ones, whereas the area of the tubes is only 2.705 times greater; the weights of air are therefore 2.5 per cent greater than might have been anticipated.

Table 9 gives analogous data for boilers Nos. 2 and 4 at 700° F. The data show that through an area 1.62 times as great, 1.84 times as much air flows, or 13 per cent more air than might have been anticipated.

It is plain that the number of sizes of tubes used was insufficient for definite deductions. Also it should be borne in mind that the air when it entered the boiler tubes from the electric furnace was not quiescent, but was moving more or less rapidly. The velocity of approach of the air as a body toward the tube sheet, therefore, was added to the velocity that there would have been in the entrances to the tubes if the furnace had been of very large volume. It is evident that the larger the tubes the greater the additional velocity due to the velocity of approach. As an extreme case, one can imagine the cross sections of the ten boiler tubes amounting to a larger area than the combined area of all the holes in the nearest asbestos-slate radiation screen. Possibly in such case the velocity of approach might increase greatly the velocity of entrance into the tubes. As a matter of fact, the combined area of the tubes of boiler No. 4 was not far from that of the holes in the last screen. However, the above reasoning is presented mostly as a suggestion of the pitfalls for which one must be on the watch in similar investigations.

TABLE 6.—*Relation between flue diameters and weights of air passing through flues, with constant pressure drops.*

BOILERS NOS. 1 AND 2.

Pressure drop through boilers (inches of water).	Weight of air flowing through tubes per second, with all initial temperatures 900° F. (pounds).		Ratios of weights of air passing through boilers Nos. 2 and 1 Column 3 Column 2
	Boiler No. 1.	Boiler No. 2.	
1	2	3	4
2	0.005	0.0077	1.44
4	.007	.015	1.50
6	.0084	.0125	1.49
8	.0095	.014	1.47
10	.0101	.015	1.48

Ratio of areas of boiler flues=boiler No. 2÷boiler No. 1=1.72.

TABLE 7.—*Relation between flue diameters and weights of air passing through flues, with constant pressure drops.*

BOILERS NOS. 1 AND 4.

Pressure drop through boilers (inches of water).	Weight of air flowing through tubes per second, with all initial temperatures 900° F. (pounds).		Ratios of weights of air passing through boilers Nos. 4 and 1 Column 3 Column 2
	Boiler No. 1.	Boiler No. 4.	
1	2	3	4
2	0.0050	0.0111	2.22
4	.0070	.0155	2.22
6	.0084	.0183	2.18
8	.0095	.0208	2.19
10	.0101	.0230	2.28
12	.0113	.0250	2.21
14	.0119	.0268	2.25
Average			2.22

Ratio of areas of boiler flues= boiler No. 4÷boiler No. 1=2.705.

TABLE 8.—*Relation between flue diameters and weights of air passing through flues, with constant pressure drops.*

BOILERS NOS. 1 AND 4.

Pressure drop through boilers (inches of water).	Weight of air flowing through tubes per second, with all initial temperatures 700° F. (pounds).		Ratios of weights of air passing through boilers Nos. 4 and 1 Column 3 Column 2
	Boiler No. 1.	Boiler No. 4.	
1	2	3	4
2	0.0049	0.0139	2.84
4	.0062	.0177	2.86
6	.0073	.0203	2.78
8	.0084	.0228	2.72
10	.0092	.0250	2.72
12	.0099	.0272	2.75
14	.0106	.0290	2.74
Average			2.77

Ratio of areas of boiler flues= boiler No. 4÷boiler No. 1=2.705.

TABLE 9.—*Relation between flue diameters and weights of air passing through flues, with constant pressure drops.*

BOILERS NOS. 2 AND 4.

Pressure drop through boilers (inches of water).	Weight of air flowing through tubes per second, with all initial temperatures 700° F. (pounds).		Ratios of weights of air passing through boilers Nos. 4 and 2 Column 3 Column 2
	Boiler No. 2.	Boiler No. 4.	
1	2	3	4
2	0.0074	0.0139	1.88
4	.0097	.0177	1.82
6	.0113	.0203	1.80
8	.0128	.0228	1.78
10	.0137	.0250	1.83
12	.0145	.0272	1.87
14	.0153	.0290	1.90
Average			1.84

Ratio of areas of boiler flues= boiler No. 4÷boiler No. 2=1.62.

RELATION OF THE LENGTH OF THE FLUES TO THE WEIGHT OF THE AIR PASSING THROUGH WHEN THE DIAMETER OF THE FLUES IS CONSTANT.

Comparison of the lower sets of curves of figures 17 and 19 shows that a given pressure drop pushes nearly the same weight of air through the 16-inch as through the 8-inch boiler. At first this fact may, perhaps, be surprising. Since the length of the flues in boiler No. 3 is double that of the flues in boiler No. 1, one would expect that the resistance to air flow of boiler No. 3 would be about twice the resistance of boiler No. 1. Probably the resistance would be nearly twice as great if the flow of air were isothermal.

The results of experiments by the authors on the flow of air through beds of shot show that with constant pressure drop the weight of air passing through a bed is inversely proportional to the thickness of the bed.^a In those experiments the authors took the simplest case of the problem, that of isothermal flow, in order to avoid complications in deducing the fundamental principles.

In the experiments with small boilers, however, the flow is not isothermal, because heat is being absorbed by the flues and the temperature of the air drops as the air flows through the tubes. This lowering of the temperature of the flowing air reduces its volume and its velocity. As the frictional resistance to the flow of gas through a cylindrical tube is proportional to the density and to the square of the velocity, it is easy to realize how the resistance of each successive portion of the boiler flue becomes less.

The relative resistances of the first 8 inches and the last 8 inches of a flue of boiler No. 3 can be approximately figured as follows:

Assuming an initial temperature of 900° F. and a temperature drop through the tube similar to that shown by the lower curve of figure 16, one finds the average temperature of the air while passing through the first 8 inches to be about 630° F., or 1,090° F. absolute, and about 320° F., or 780° F. absolute, while passing through the last 8 inches of the flues. The average volumes and, therefore, the average velocities of the air in the two portions of the flues are proportional to these absolute temperatures.

The relative frictional resistances in the two halves of the flue, so far as the velocity factor is concerned, are then approximately as 1,090² to 780². But the friction is also proportional to the density, which is inversely proportional to the absolute temperature. Therefore, one may say that the friction in the first half is to that in the second half of the tube as 1,090² ÷ 1,090 is to 780² ÷ 780, or as 1,090 is to 780; 780 is 71.5 per cent of 1,090. The above reasoning takes into account only the effect due to the cooling of the air. Undoubt-

^a See Bull. 21, Bureau of Mines, The significance of drafts in steam-boiler practice, p. 31.

edly, at the entrance of the air into the tubes there is some kind of "vena contracta" effect, which further increases the resistance of the first half of the flues; there is also a loss of head on the sudden expansion of the air in leaving the tubes.

If (the reader will please bear in mind the if) the total resistance to the flow of air is inversely proportional to the weight of air flowing through the flues with any given pressure drop, the resistances of the 8-inch and the 16-inch flues can be approximated by means of figures 17 and 19, in the following manner:

With a 4-inch pressure drop and an initial temperature of 900° F. the weight of air flowing through the 8-inch boiler, according to figure 17, is 0.007 pound per second; under the same conditions the weight flowing through the 16-inch boiler, according to figure 19, is about 0.0058 pound. The ratio of the two weights is $0.007 \div 0.0058 = 1.20$; hence, that if the total resistance of the 8-inch boiler is 1 the total resistance of the 16-inch boiler is 1.20.

TABLE 10.—*Comparative weights of air flowing through two boilers with flues of the same diameter but of different lengths.*

[Initial temperature, 900° F.]

Pressure drop through boiler (inches water).	Weight of air flowing through—		Ratios of the weights of air.
	Boiler No. 1 (flues 8 inches long).	Boiler No. 3 (flues 16 inches long).	Boiler No. 1. Boiler No. 3.
	<i>Pounds.</i>	<i>Pounds.</i>	
2	0.0050	0.0040	1.25
4	.0070	.0059	1.20
6	.0084	.0070	1.19
8	.0095	.0080	1.19
10	.0104	.0090	1.15
12	.0112	.0101	1.10

These ratios have been figured for six different pressure drops, the approximate values of the weights of air as obtained from figures 17 and 19 being used, and are given in Table 10. As the table shows, the ratio decreases when the pressure drop increases. This fact seems to indicate that as the pressure drop increases the resistance due to the length of the flue decreases, while the resistance at the entrance of the flues remains about the same or perhaps increases, the result being that the total resistance of the 16-inch flues gradually approaches that of the 8-inch flues.

The following deduction may be drawn:

In the flow of gas through boiler flues much of the resistance seems to be at the entrance of the gas into the flue. Increasing the length of the flue increases the resistance but little.

RELATION OF THE RATE OF HEAT ABSORPTION TO THE PRESSURE DROP WHEN THE INITIAL TEMPERATURE REMAINS CONSTANT.

Figures 21 to 25 show the relation between the effective pressure drop through the boilers and the rate of heat absorption.

Each figure gives this relation for one constant initial temperature, and the curve for each boiler is indicated by number. By effective

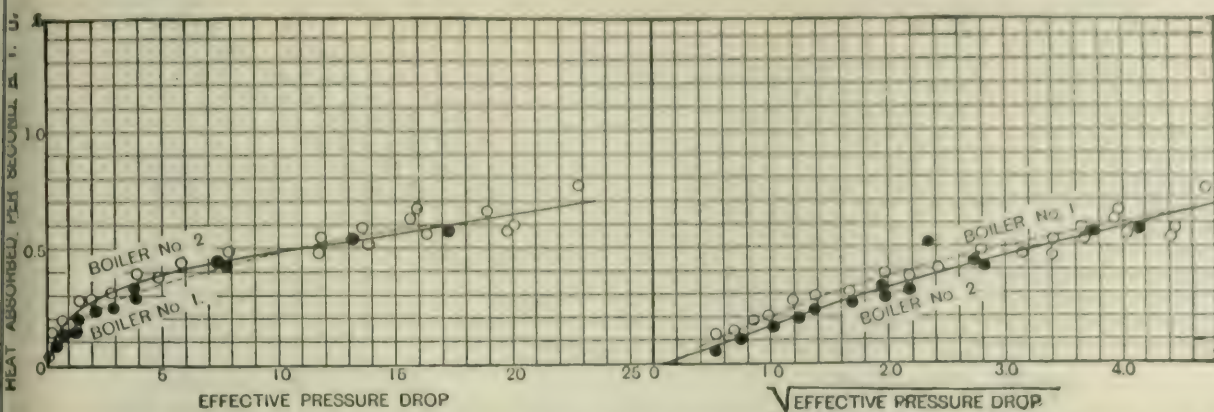


FIGURE 21.—Heat (B. t. u.) absorbed per second by each of two boilers, as a function of both the pressure drops through the boilers and the square roots of these pressure drops. All points for initial temperatures of 500° F.

pressure drop is meant the difference between the air pressures at the two ends of the boiler. In the left half of each figure the plain pressure drops are used as abscissæ, in the right-hand portion the square roots of the effective pressure drops are so used. The indi-

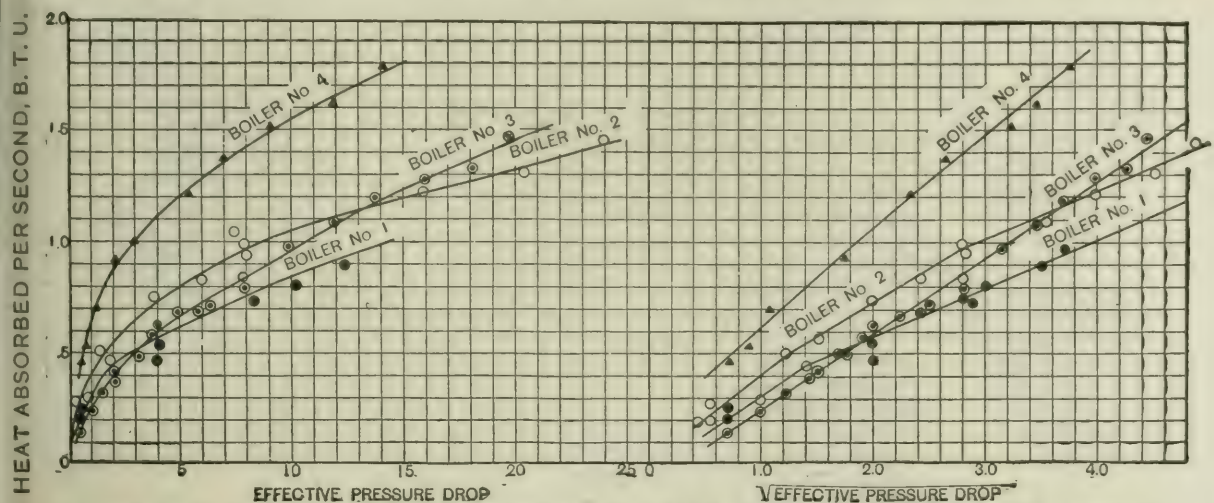


FIGURE 22.—Heat (B. t. u.) absorbed per second by each of four boilers, as a function of both the pressure drops through the boilers and the square roots of these pressure drops. All points for initial temperatures of 700° F.

cation of the two halves is much the same, the only difference being that the curves on the right are nearly straight lines. The object of plating the curves on the right was to show that since the weight of air passing through a boiler is nearly proportional to the square root of the pressure drop, the rate of heat absorption is also nearly proportional to the square root of the pressure drop.

The curves in the left halves of the figures are similar in shape to the lower curves of figures 17 to 20, which show that as the

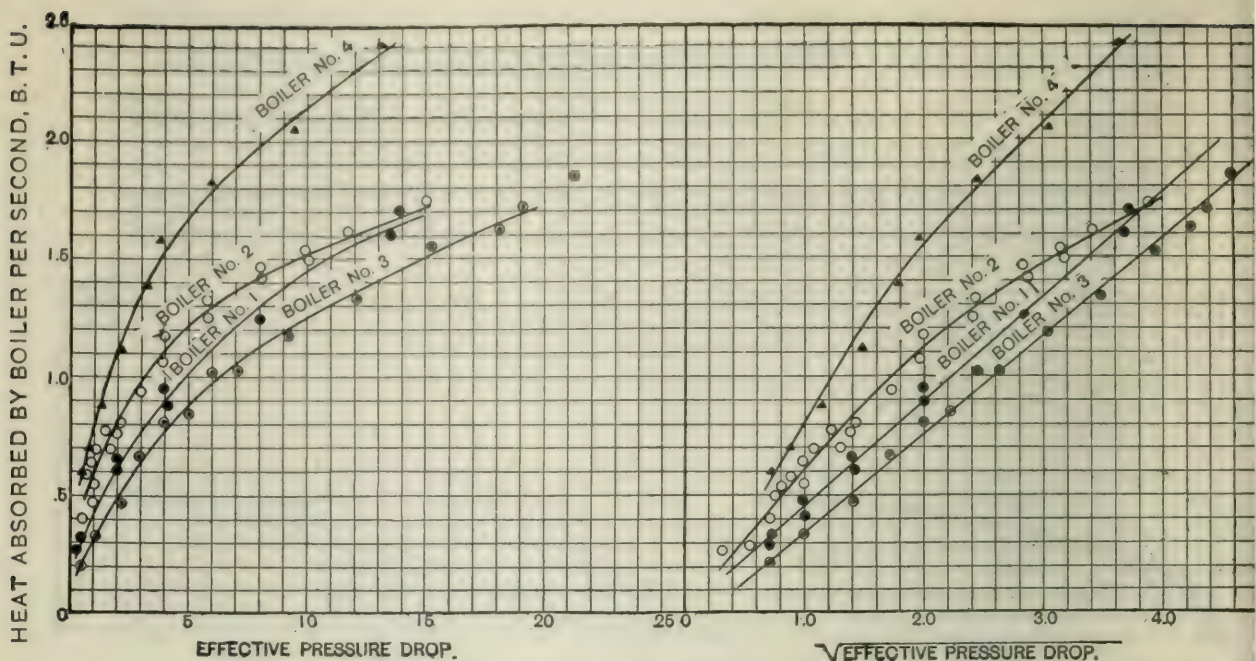


FIGURE 23.—Heat (B. t. u.) absorbed per second by each of four boilers, as a function of both the pressure drops through the boilers and the square roots of these pressure drops. All points for initial temperatures of 900° F.

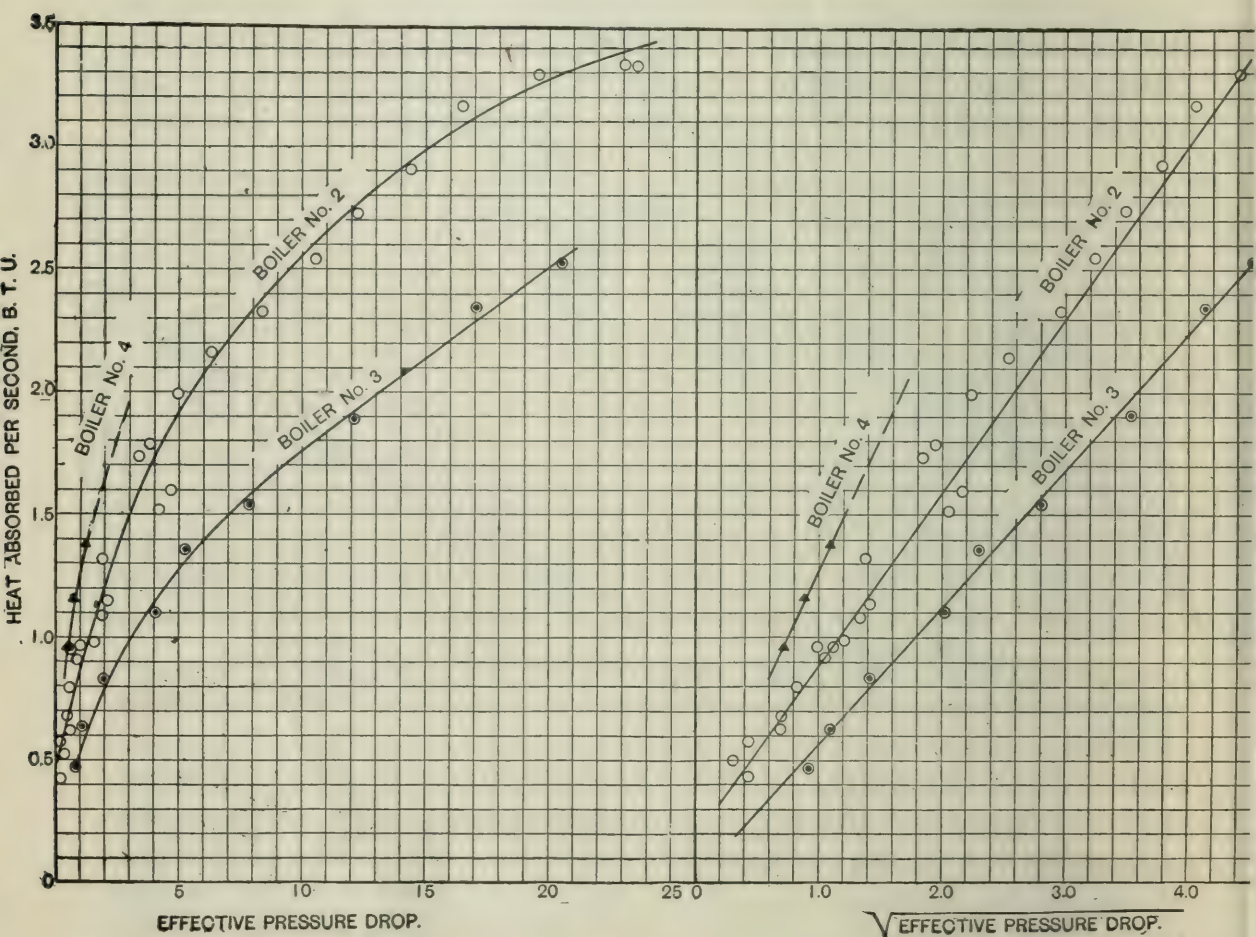


FIGURE 24.—Heat (B. t. u.) absorbed per second by each of four boilers, as a function of both the pressure drops through the boilers and the square roots of these pressure drops. All points for initial temperatures of 1,200° F.

pressure drop increases the quantity of heat absorbed rises rapidly at first and then more slowly. The similarity of the two sets of curves must necessarily follow from the fact that the weight of air passing through a boiler and the rate of heat absorption increase and decrease together, as stated on page 48.

On comparison of figures 21 to 25, in the order of the various initial temperatures given, it will be seen that at the higher temperatures

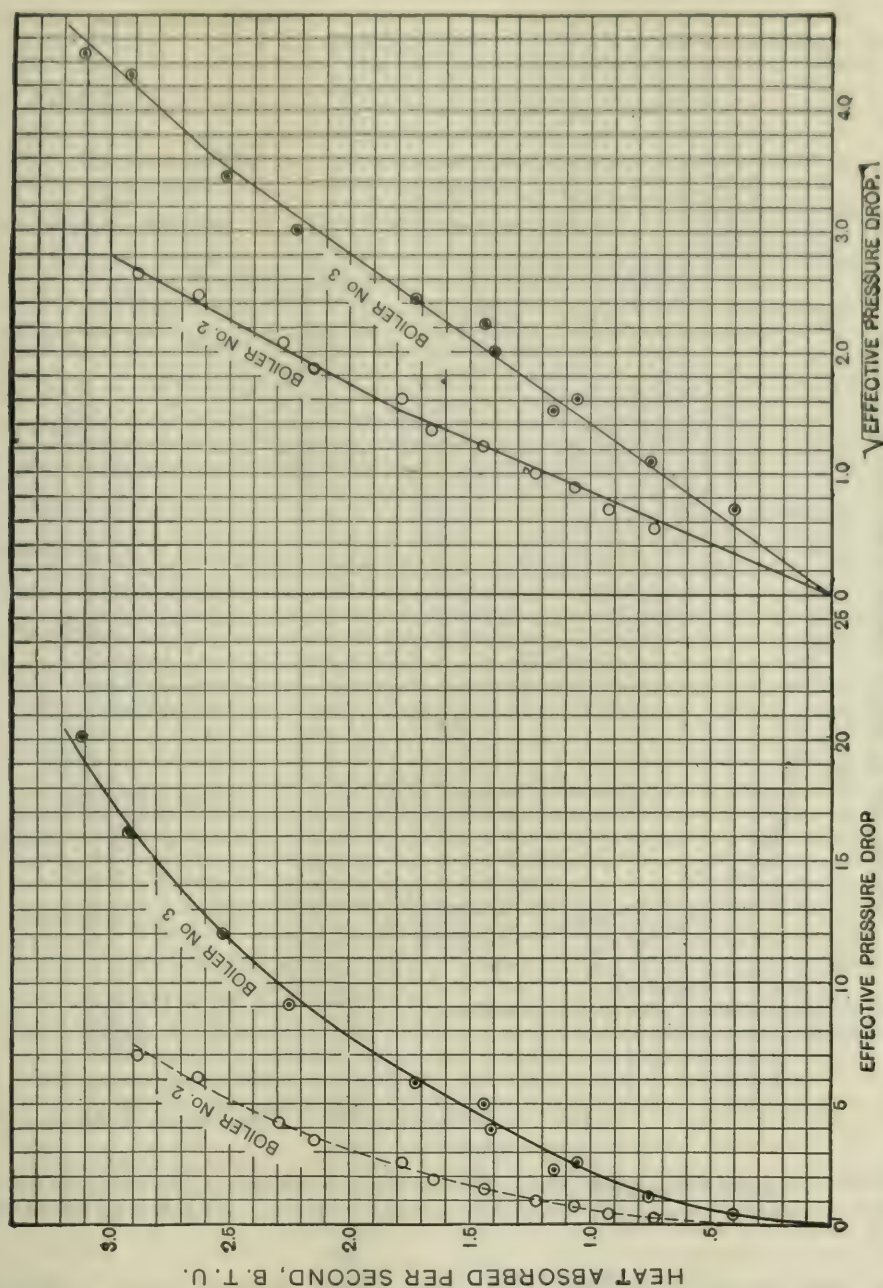


FIGURE 25.—Heat (B. t. u.) absorbed per second by each of three boilers, as a function of both the pressure drops through the boilers and the square roots of these pressure drops. All points for initial temperature of 1,500° F.

the rate of heat absorption increases much more rapidly for a given increase in pressure drop than for a similar increase of the pressure drop at lower temperatures. This is partly due to the fact that at the higher initial temperatures the same pressure drop pushes through a larger weight of air; it is chiefly due, however, to the fact that 1 pound of air holds more heat available to the boiler

at a high than at a low initial temperature. Therefore, the boiler absorbs more heat from air at high than from air at low initial temperatures.

A comparison of curves of the same initial temperature for the four boilers shows that up to about 900° F. all curves except the one for boiler No. 4 are close together; above that temperature the curve of boiler No. 2 gains rapidly over that of boiler No. 3.

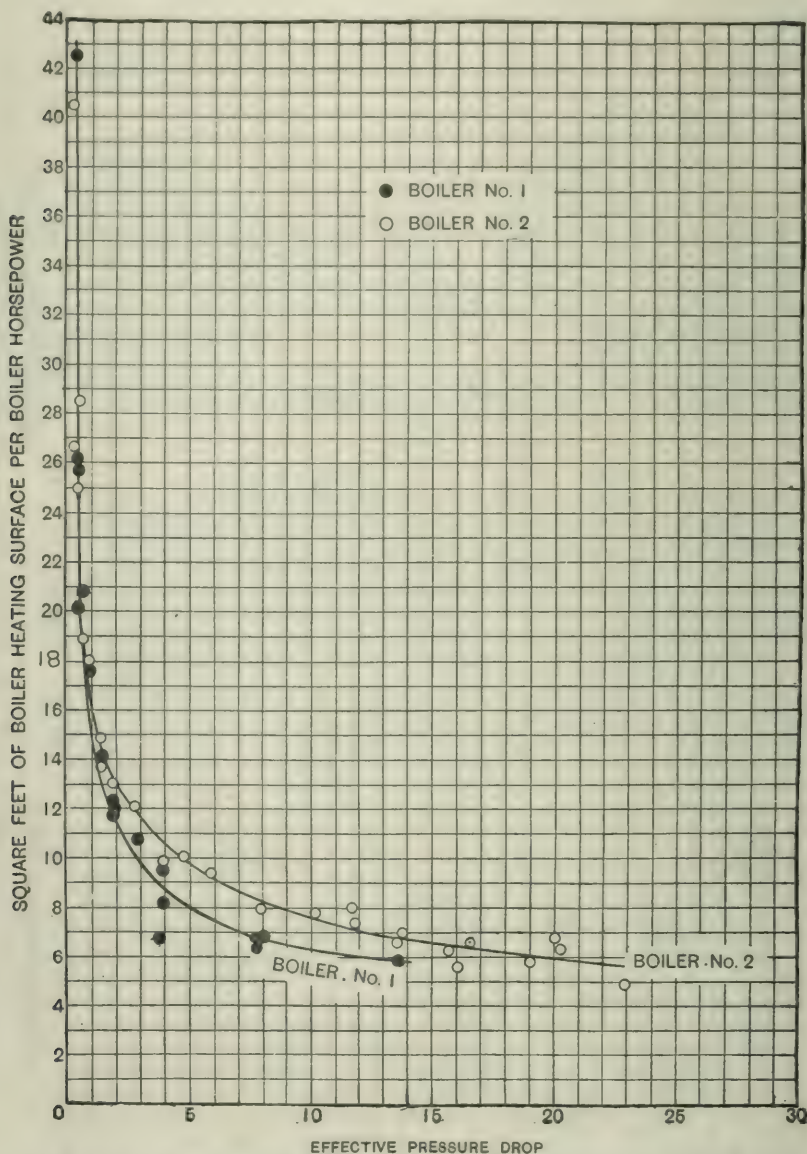


FIGURE 26.—Influence of the pressure drops of air, while passing through each of two boilers, on the number of square feet of heating surface required to deliver 1 boiler horsepower. All points for initial temperatures 500° F.

This gain is due to the fact that the same pressure drop pushes a much larger quantity of air through boiler No. 2 than through boiler No. 3, therefore the former absorbs more heat than the latter, since its efficiency is only about one-third lower.

In regard to the curves at the right-hand side of the figures, one can state that when the initial temperature remains constant the boiler capacity varies in almost direct proportion to the square root of the pressure drop.

RELATION OF THE NUMBER OF SQUARE FEET OF HEATING PLATE (SURFACE) REQUIRED PER BOILER HORSEPOWER TO THE EFFECTIVE PRESSURE DROP.

Figures 26 and 28 bring out the effect of variations in the pressure drop of air, while passing through each of two and three boilers, on the square feet of heating surface required per boiler horsepower, for initial air temperatures of 500° and 700° F., respectively. Figures 27 and 29 are analogous charts, which bring out the same effect for variations in the square roots of the same pressure drops. Figures 30 to 32 are for initial temperatures of 900°, 1,200°, and 1,500° F., respectively; for each of these temperatures it was possible to consolidate the two showings into one figure. Each of the curves on these charts is for the boiler designated by the number.

All of the curves of this group of figures (26-32) show that as the pressure drop increases the area of the heating surface required per boiler horsepower decreases; at first the decrease in the area of the heating surface for each increase of pressure drop is rapid, but after medium values of the

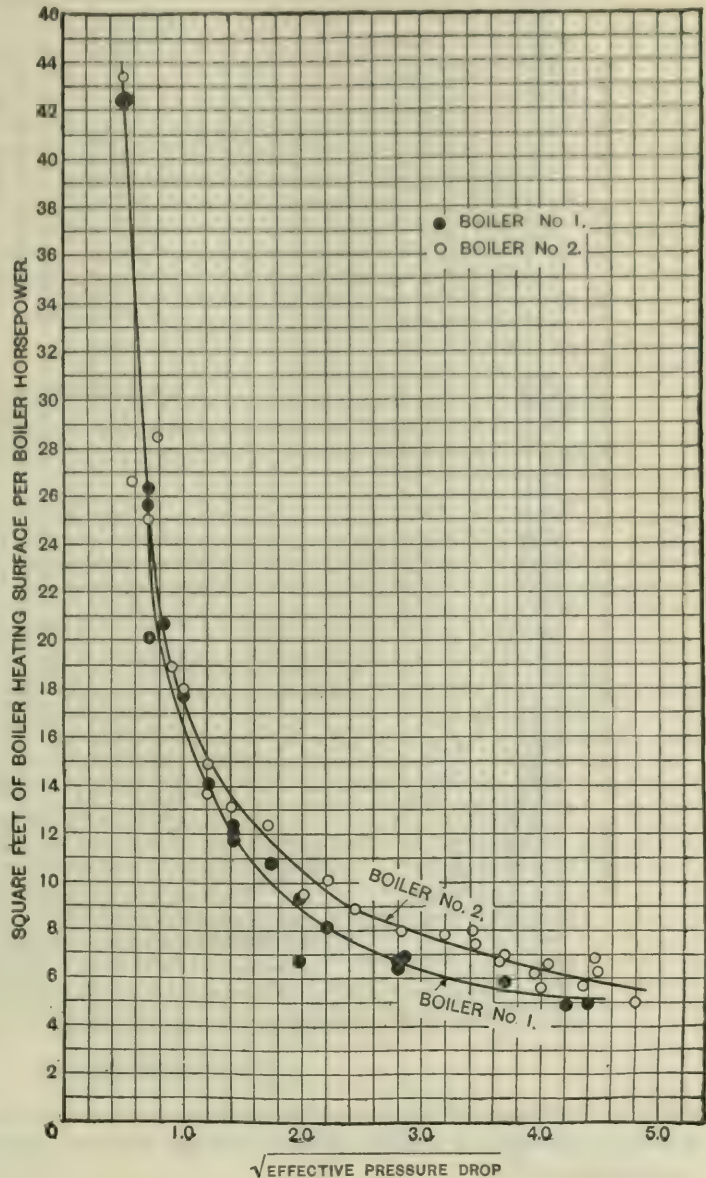


FIGURE 27.—Influence of the square roots of the pressure drops of air, while passing through each of two boilers, on the number of square feet of heating surface required to deliver 1 boiler horsepower. All points for initial temperatures 500° F.

pressure drop are passed the decrease becomes less and less. It will be noticed that the abrupt bend in the curve, with any initial temperature and almost any boiler, occurs with about the same pressure drop. The area of heating surface per boiler horsepower near the bend in the curves decreases with the increase in the initial temperatures. Thus, for boiler No. 2, with an initial temperature

of 500° F., the heating surface required per boiler horsepower is about 10 square feet, whereas with an initial temperature of 1,500° F. it is only about 1.75 square feet. A still higher initial temperature would further decrease this value, although by a very small quantity.

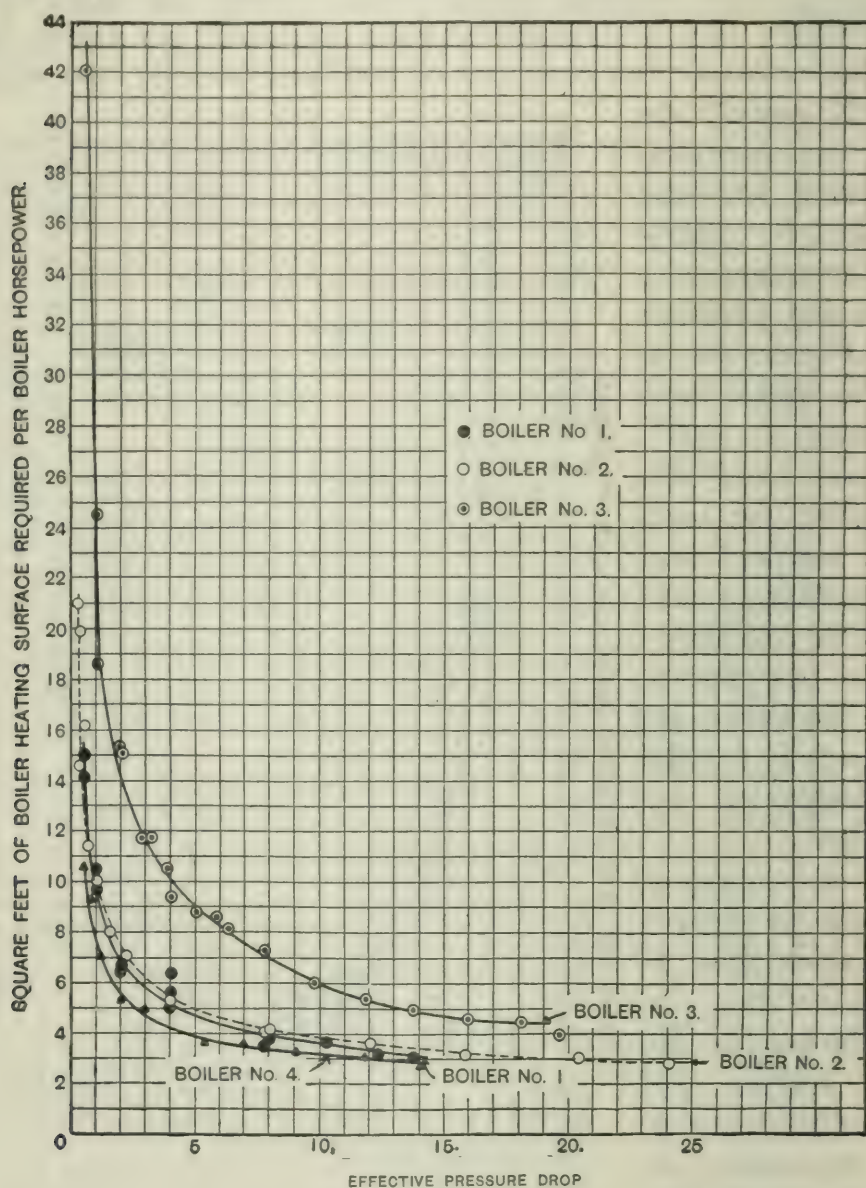


FIGURE 28.—Influence of the pressure drops of air, while passing through each of four boilers, on the number of square feet of heating surface required to deliver 1 boiler horsepower. All points for initial temperatures 700° F.

The exact relation between the heating surface required and the pressure drop seems to be decidedly complicated. Perhaps the simplest roughly approximate law can be deduced from the charts having the square roots of the pressure drops for abscissæ.

Thus, in figure 29 (for boiler No. 2), when the heating-surface area per boiler horsepower is 20 square feet, the square root of the pressure drop is about 0.5; to reduce the heating-surface area one-half, the square root of the pressure drop must be about 1.1, or about

twice as much as in the first case. If the heating surface is to be reduced to 5 square feet, or one-fourth that of the first case, the square root of the pressure drop must be increased to about 2.2, or about four times as much as in the first case.

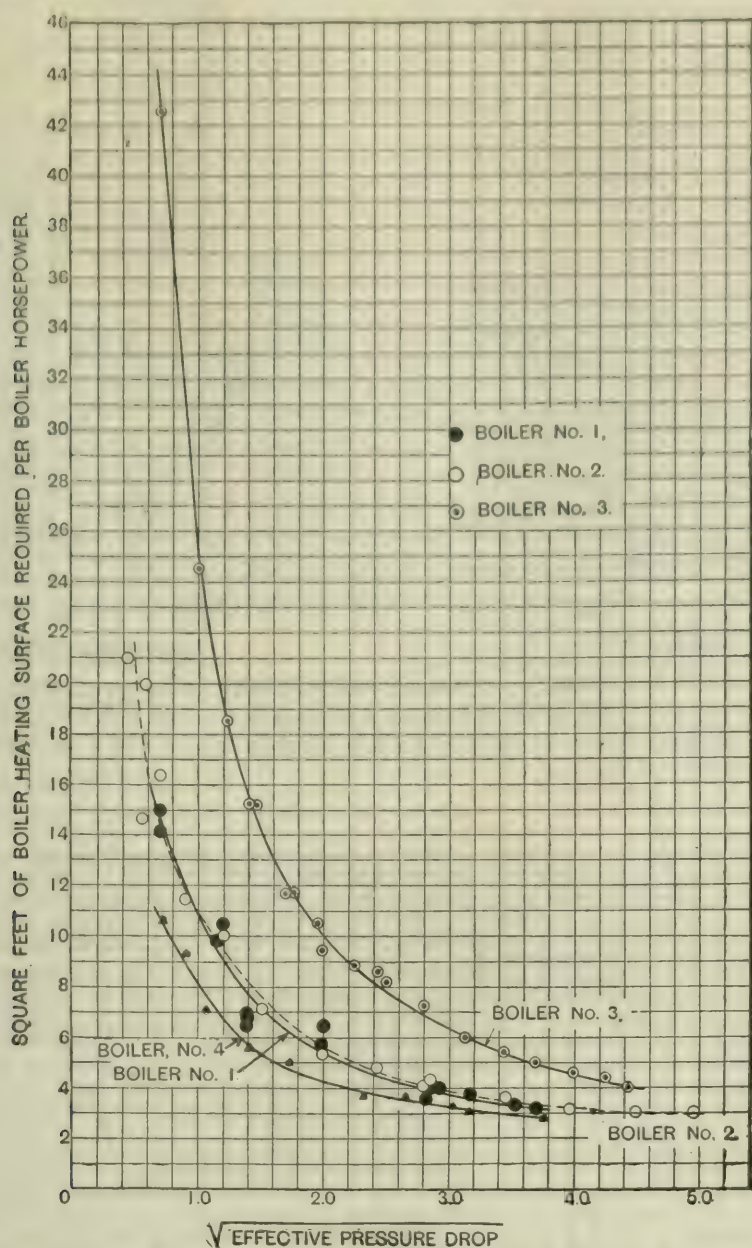


FIGURE 29.—Influence of the square roots of the pressure drops of air, while passing through each of three boilers, on the number of square feet of heating surface required to deliver 1 boiler horsepower. All points for initial temperatures 700° F.

In the curves for boiler No. 3 the heating-surface area decreases in somewhat faster ratio than accords with this law of inverse proportion to the square root of the pressure drop. Thus, when the heating surface is 20 square feet, the square root of the pressure drop is about 1.15; when the heating surface is reduced to 10 square feet the square root of the pressure drop is 2, or not quite twice as much as in the first case. To reduce the heating surface to 5 square feet

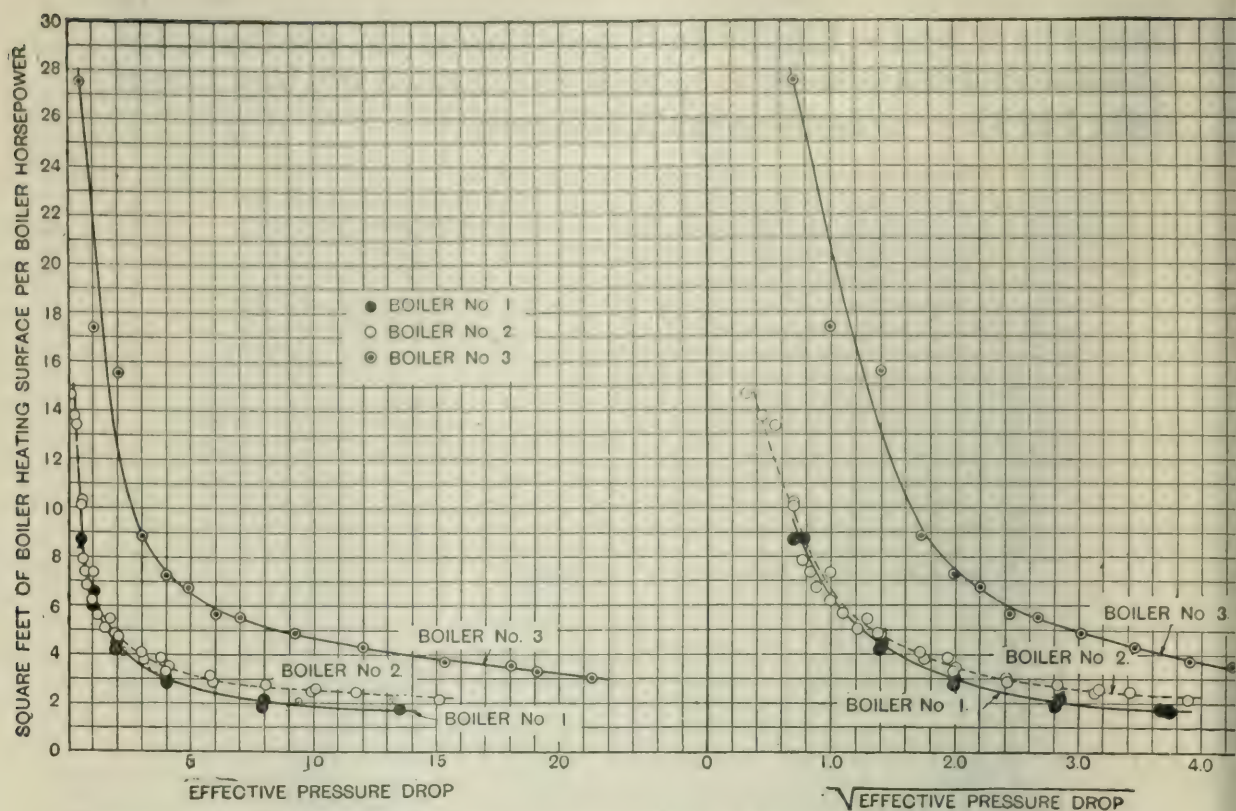


FIGURE 30.—Area, in square feet, of heating surface required to deliver 1 boiler horsepower in each of three boilers function of both the pressure drops of air while passing through the boilers and the square roots of these pressure drops. The curves for boiler No. 4 would be so close to those for Nos. 1 and 2 that they were not plotted. Points for initial temperatures 900° F.

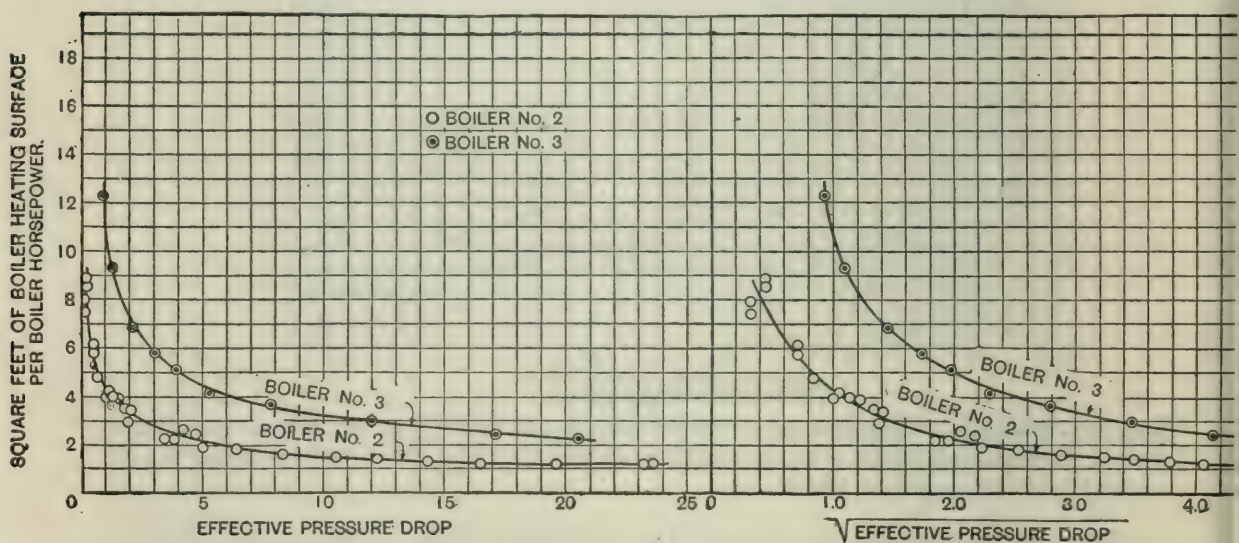


FIGURE 31.—Area, in square feet, of heating surface required to deliver 1 boiler horsepower, in each of two boilers function of both the pressure drops of air while passing through the boilers and the square roots of these pressure drops. Curves for boiler No. 4 were not drawn because there were only three points for each, all at very low pressure drops, and because the curves would have fallen only a little under those for boiler No. 2. All points for initial temperatures 1,200° F.

the square root of the pressure drop needs be increased only to 3.6 instead of 4.6 (4×1.15). With such low temperatures as 700°F. , and for this particular boiler, it would take very high pressure drops to reduce below 3 square feet the heating surface required per boiler horsepower. However, with temperatures of $1,500^{\circ}\text{F.}$, and with shorter-tubed boilers, the heating surface can be reduced to less than $1\frac{1}{2}$ square feet without the need of excessive pressure drops; this conclusion is supported by the curve of boiler No. 2 in figure 32.

The amount of reduction of the heating surface should always be discussed as a percentage and not as so many square feet. A much higher pressure drop is necessary to reduce the heating surface 1 square foot from an original 4 square feet than to reduce it 1 square foot from an original 10 square feet. It should not be forgotten, however, that in the first case the rate of working the boiler is increased by 33 per cent through the reduction in the heating surface, whereas in the second case the rate of working is increased by only 11 per cent.

There is one deduction which can be drawn conclusively from these groups of figures, and that is the number of square feet of heating surface required per boiler horsepower decreases as the initial temperature of the gases and the effective pressure drop increases. There

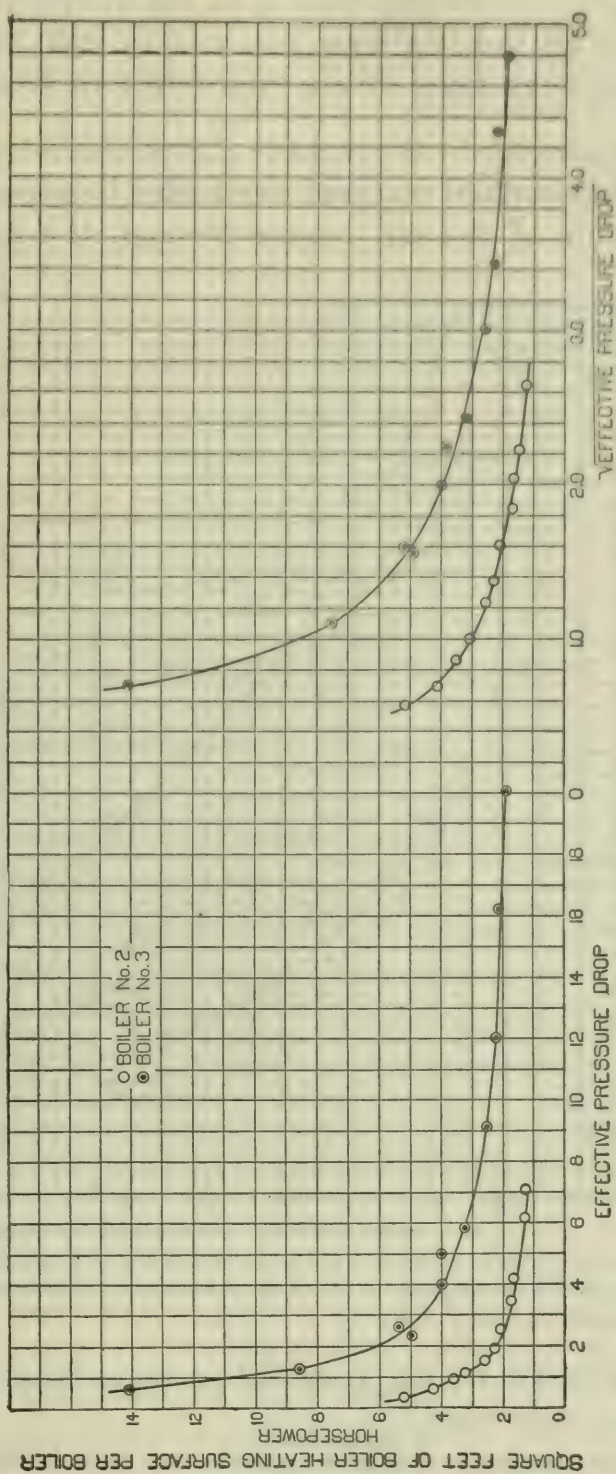


Figure 32.—Area, in square feet, of heating surface required to deliver 1 boiler horsepower, in each of two boilers, as a function of both the pressure drops of air while passing through the boilers and the square roots of these pressure drops. All points for initial temperatures $1,500^{\circ}\text{F.}$

is just as much logic in saying that the heating surface necessary to produce 1 boiler horsepower is 20, or 3 square feet, as in saying that it is 10 square feet.

RELATION BETWEEN THE AREA OF HEATING SURFACE REQUIRED PER BOILER HORSEPOWER AND THE INITIAL TEMPERATURE WHEN THE PRESSURE DROP OF AIR THROUGH THE BOILERS REMAINS CONSTANT.

Figures 33 to 36, inclusive, give the relation between the area of heating surface required per boiler horsepower and the initial temperature of the air passing through the boiler. Each figure gives

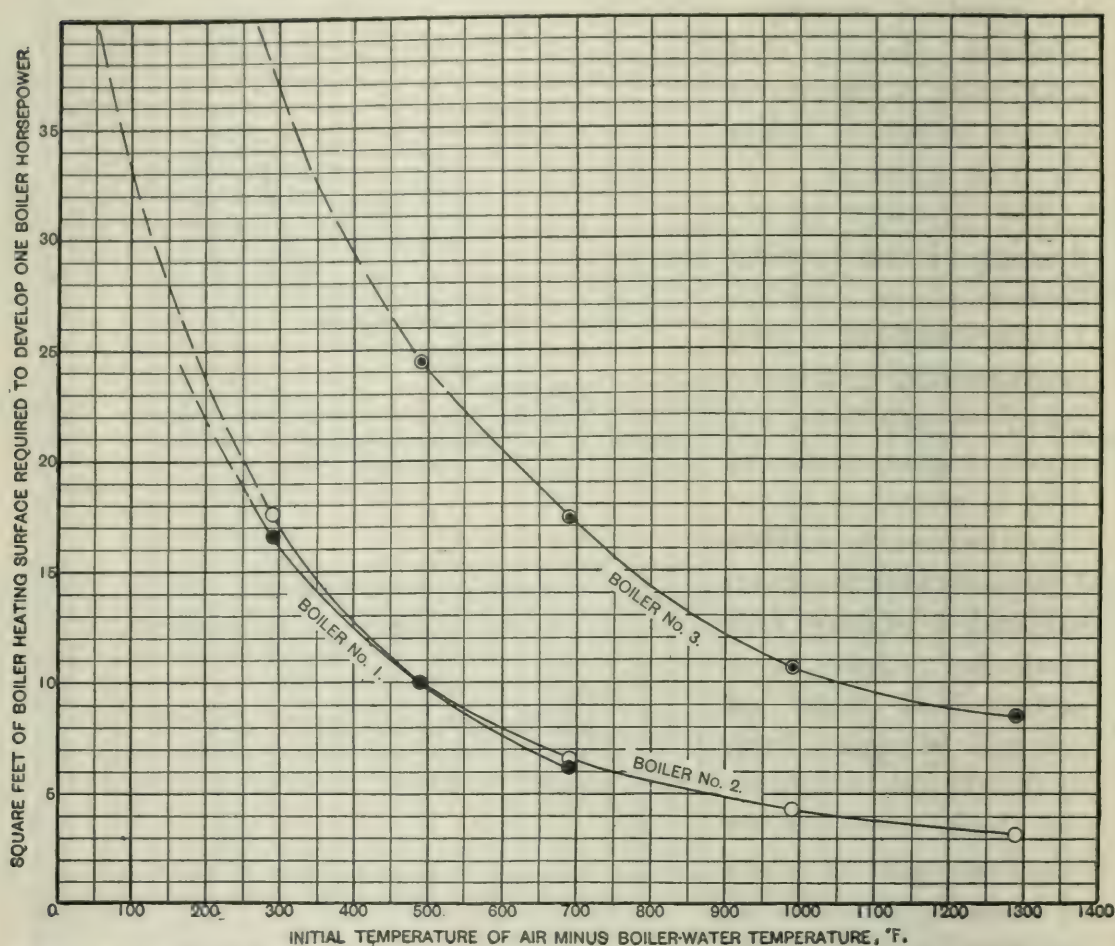


FIGURE 33.—Area, in square feet, of heating surface required to develop 1 boiler horsepower, as a function of the initial air temperature minus the boiler-water temperature, when the pressure drop of the air while passing through the boilers always equals 1 inch of water.

this relation for one constant pressure drop, and each curve is for the boiler designated.

The abscissæ of all the charts are the differences between the initial temperatures of the air and the temperatures of the steam in the boilers. Since the latter temperatures are almost constant, the variation of the difference is about the same as the variation of the initial temperature. The curves have been worked out for four different pressure drops, namely, 1 inch, 4 inches, 9 inches, and 16 inches of

water; these figures were taken on account of being perfect squares, their square roots being 1, 2, 3, and 4.

All the curves appear to be asymptotic at the left to the vertical line of zero temperature difference, and at the foot of the charts to the horizontal line of zero heating surface. These relations seem to be required, for if the temperature difference were nearly zero the area of heating surface required to develop a boiler horsepower would have to be infinitely large, and if the heating surface were reduced nearly to zero the temperature difference would have to be infinitely

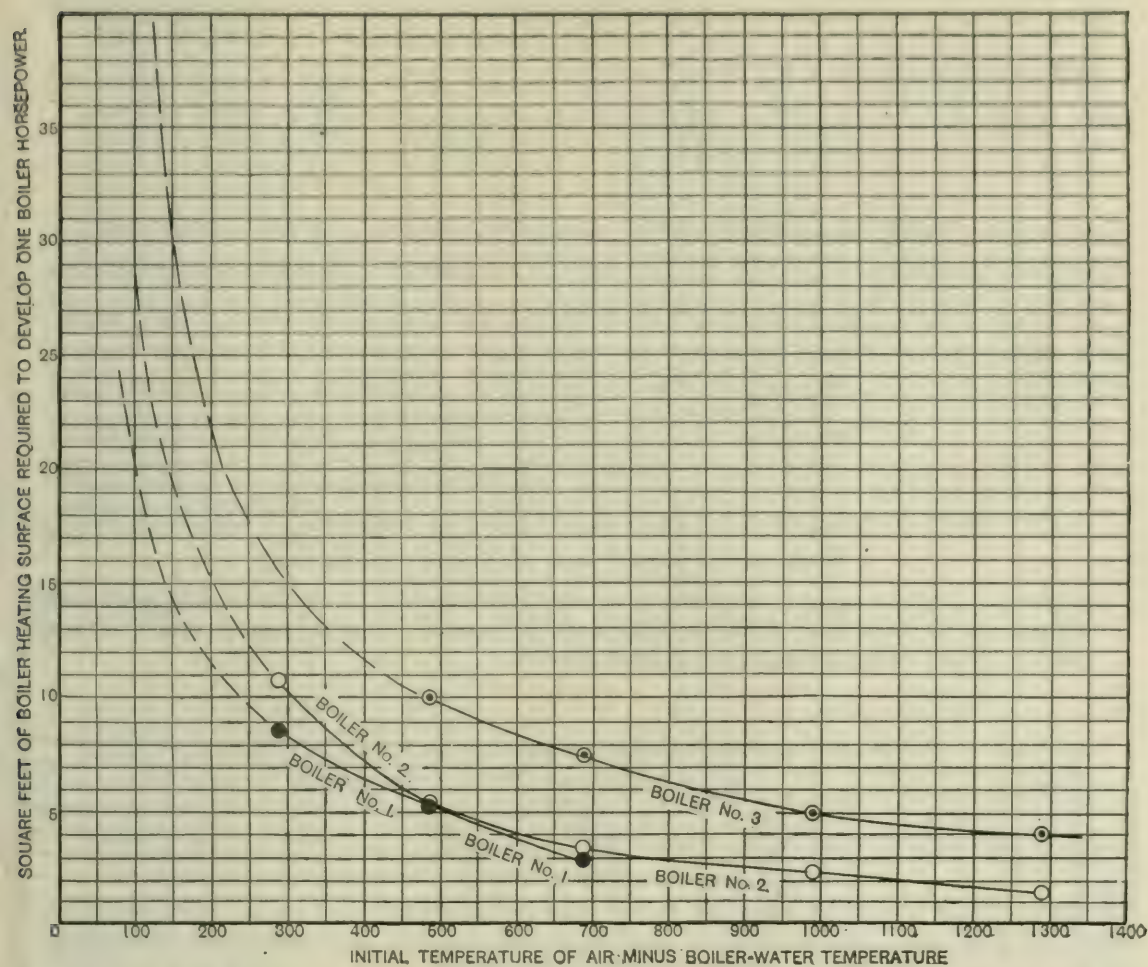


FIGURE 34.—Area, in square feet, of heating surface required to develop 1 boiler horsepower, as a function of the initial air temperature minus the boiler-water temperature, when the pressure drop of air while passing through the boilers always equals 4 inches of water.

high. Between these two limits the heating surface seems to decrease almost as fast as the temperature difference increases. This relation seems to hold at least for the temperatures investigated. There are reasons, however, which lead to the belief that after moderate temperatures are reached the decrease in the heating surface would be considerably less with further increases of temperature than this relation would call for. One reason why, in the case of the two short boilers, the reduction of the heating surface keeps up so well with the temperature increase is that the same pressure drop pushes

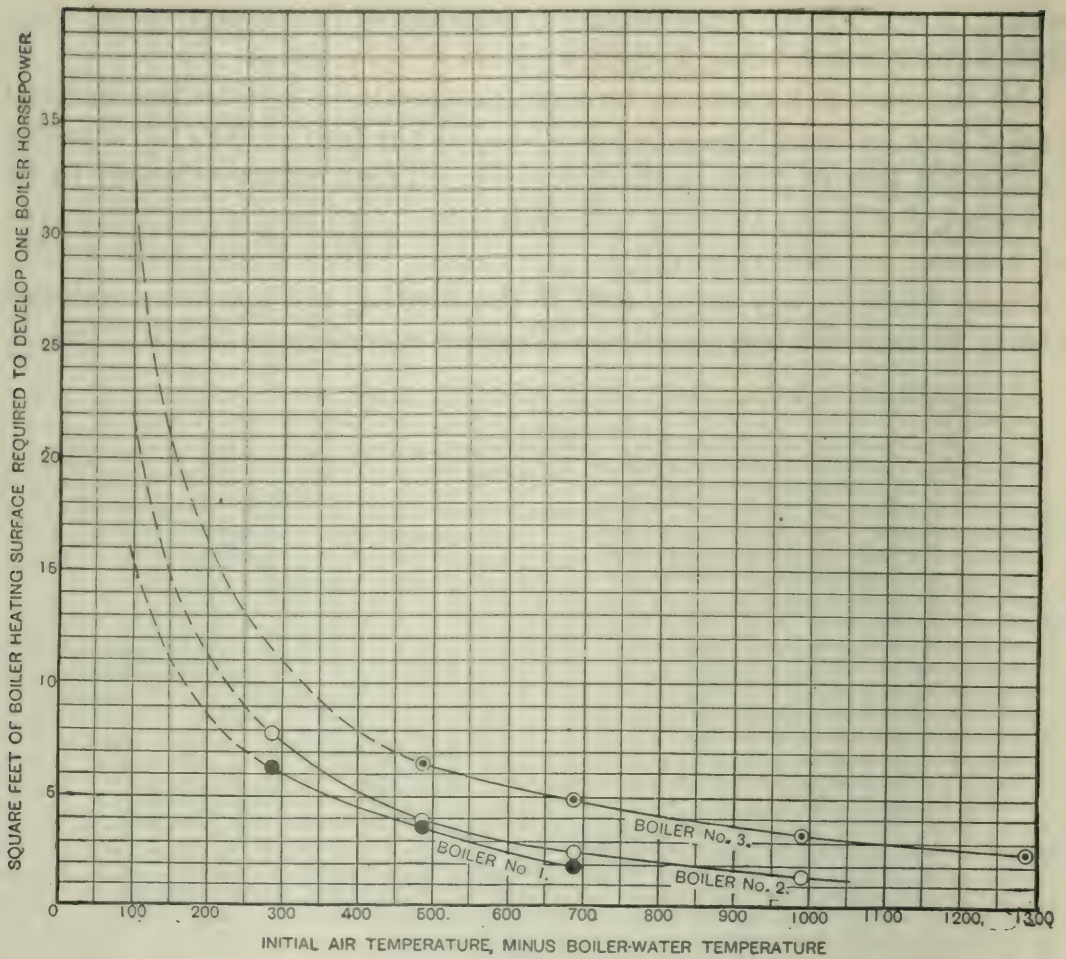


FIGURE 35.—Area, in square feet, of heating surface required to develop 1 boiler horsepower, as a function of the initial air temperature minus the boiler-water temperature, when the pressure drop of air while passing through boilers always equals 9 inches of water.

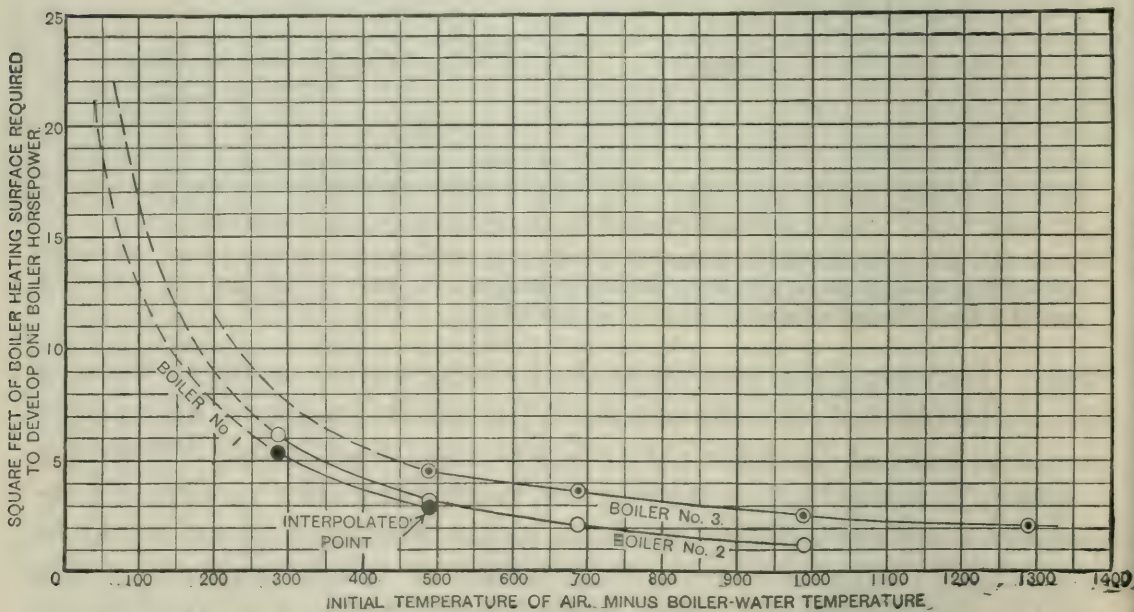


FIGURE 36.—Area, in square feet, of heating surface required to develop 1 boiler horsepower, as a function of the initial air temperature minus boiler-water temperature, when the pressure drop of air while passing through boilers always equals 16 inches of water.

through the boiler larger weights of gases when the initial temperature is higher.

Comparing boilers Nos. 1 and 2, which are of the same length but have flues of different diameters, it is interesting to note that with the same pressure drop and the same initial temperature the boiler with the small flues requires less heating surface per boiler horsepower than the boiler with the larger flues.

With low-pressure drops the values of the heating surface of boiler No. 3 are rather high as compared with boilers Nos. 1 and 2; but, as the pressure drop increases, boiler No. 3 comes close to the other two boilers, absolutely, but not in percentage.

RELATION BETWEEN THE PRESSURE DROPS AND INITIAL TEMPERATURES WHEN THE AMOUNT OF HEATING SURFACE REQUIRED PER BOILER HORSEPOWER REMAINS CONSTANT.

Figures 37 to 40, inclusive, have been devised to show the interrelations between the pressure drops of the air while passing through the boilers and the differences between the initial air and the boiler-water temperatures when the area of heating surface required per boiler horsepower remains constant. Each figure is worked out for the constant area of heating surface stated in its legend.

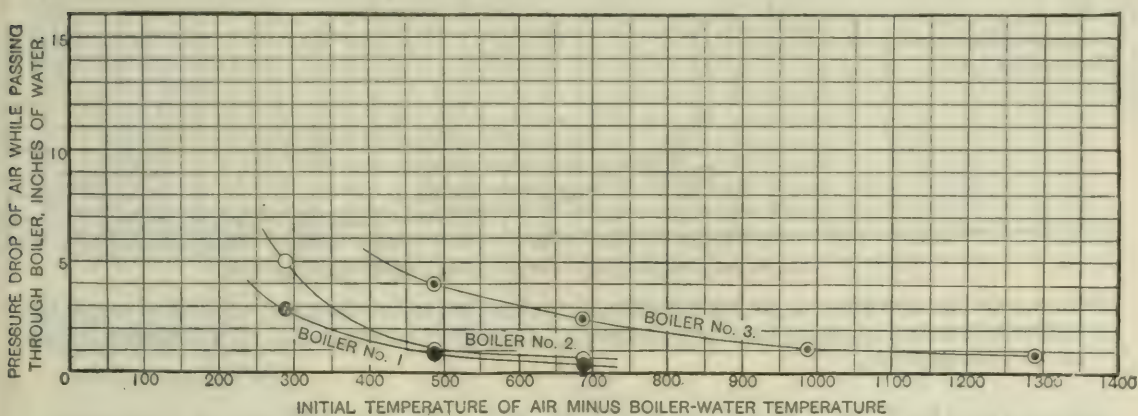


FIGURE 37.—Pressure drops of air while passing through boilers as functions of the initial air minus the boiler-water temperatures for a rate of heat absorption always requiring 10 square feet of heating surface per boiler horsepower.

The abscissæ are the differences between the initial temperatures of the air and the boiler-water temperatures. The water temperature is substantially constant, so that the variation of the difference is the same as that of the initial temperature. This relation between the pressure drops and the initial temperatures, when the rate of heat absorption remains constant, can not be exactly expressed in the form of a simple law. Perhaps it may be suggested that the product obtained by multiplying the square root of the pressure drops by the initial temperature minus that of the water in the boiler is roughly constant. However, this relation is only approximately true and will

not hold for all boilers through any range of temperatures; it holds best at the high temperatures of boiler practice.

The charts merely show that by certain combinations of the pressure drops and the initial temperatures almost any boiler can be made to work at any desired rate.

On the chart (fig. 37) based on 10 square feet of heating surface per boiler horsepower the curves of boilers Nos. 1 and 2 are rather close together; the curve for boiler No. 3 is much higher than those for the other two boilers and rises still higher on the charts based on smaller areas of heating surface. The reason why, with the same initial temperature, boiler No. 3 requires a much higher pressure drop to make its rate of heat absorption equal to the rates of the

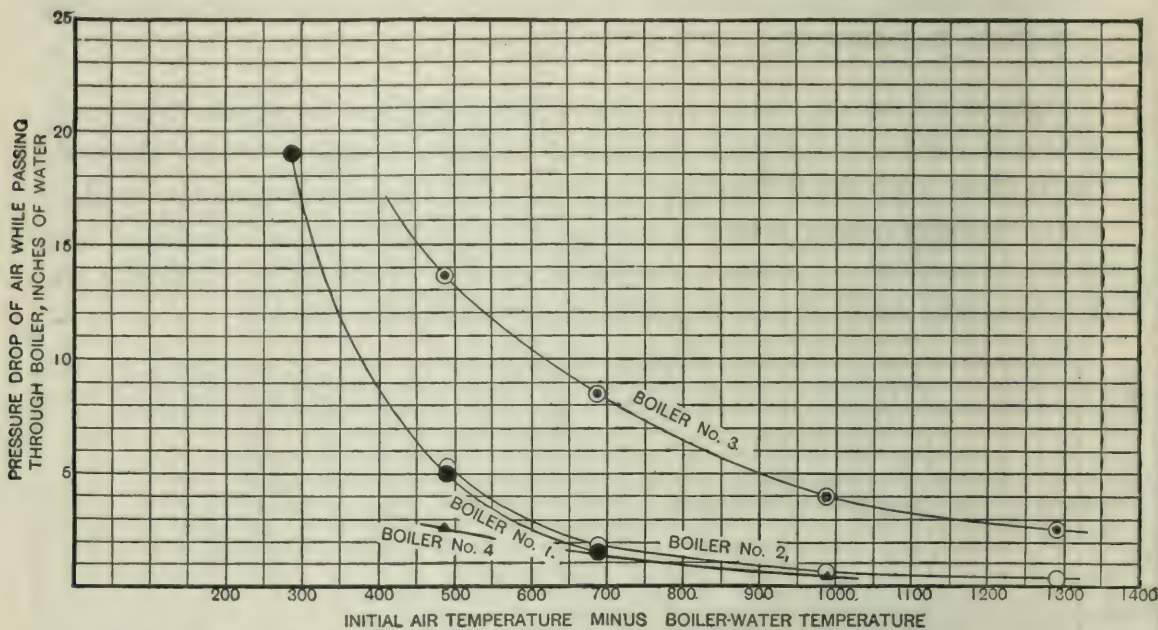


FIGURE 38.—Pressure drops of air while passing through boilers as functions of the initial air minus the boiler-water temperatures for a rate of heat absorption always requiring 5 square feet of heating surface per boiler horsepower.

other two boilers is that its resistance to the flow of gas is high compared to that of boiler No. 2; further, it has a large heating surface compared to boiler No. 1. Boiler No. 3 has twice the heating surface of boiler No. 1; therefore, in order that the heating surface of the former be worked as hard as that of the latter, nearly twice as much air must be pushed through boiler No. 3 as through boiler No. 1. Consequently nearly four times as much pressure drop is required for boiler No. 3 as for boiler No. 1. However, it should be remembered that boiler No. 3 absorbs about 10 per cent more heat from each pound of air than does boiler No. 1, which has flues of the same diameter but only half as long.

As between boilers Nos. 1 and 2, the former is preferable because a smaller pressure drop is required to work it at the same rate; moreover, boiler No. 1 absorbs about 14 per cent more heat from each

pound of air than does boiler No. 2. This advantage of smaller tubes is of extreme importance in boiler construction.

The curve for boiler No. 4 is not on figure 37 because it would have fallen too close to that for boiler No. 1, although a little below the latter. In figure 38 for the low temperatures the curve of boiler No. 4 lies below the others, but for the higher temperatures it nearly coincides with them. In figures 39 and 40 the curves for boiler No. 4 lie between those for boilers Nos. 1 and 2, probably because of an error somewhere. As was remarked before, the tests on boiler No. 4 are not nearly so reliable as those on the other three. More-

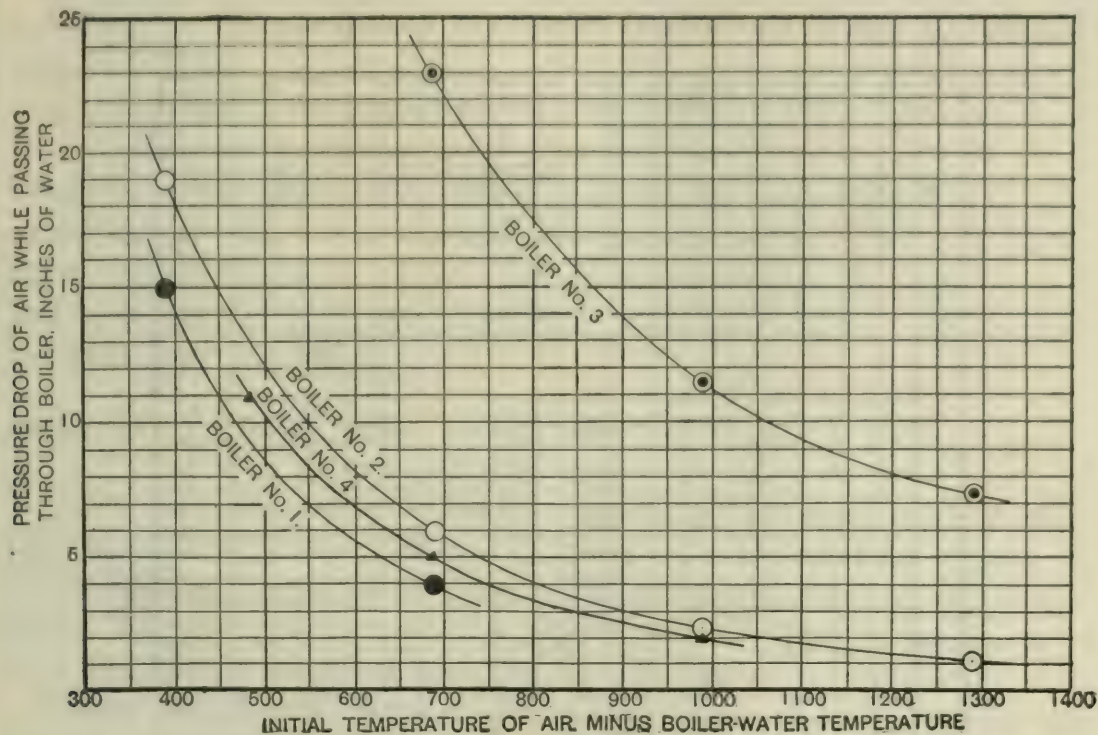


FIGURE 39.—Pressure drops of air while passing through boilers as functions of the initial air minus the boiler-water temperatures for a rate of heat absorption always requiring 3 square feet of heating surface per boiler horsepower.

over, the larger total area of the cross sections of the tubes in this boiler increased the velocity of approach of the air to the tube entrances, so that the quantity of air that passed through was larger than that due merely to the head or pressure drop through the boiler.

It is regrettable that several more sizes and lengths of tubes were not tested; nevertheless the indications of figures 32, 33, and 34, so far as they concern boilers Nos. 1, 2, and 4, probably justify to some extent the following deduction:

If, in any fire-tube boiler, the tubes are replaced by smaller ones of the same total heating surface (not the same total cross-sectional area), the same pressure drop will result in the production of about the same quantity of steam, at a higher true boiler efficiency.

THE REALITY OF A CRITICAL VELOCITY IN FLOWING FLUIDS.

At a number of places in this bulletin the rate of flow of gas or water in a tube is referred to as below or above the "critical velocity." This term "critical velocity" is not at all self-explanatory; it is that velocity at which the stream lines of flow cease rather suddenly to be parallel, and at which violent eddying begins. Prof. Osborne Reynolds, one of the first men to notice it, wrote upon it many years ago; he found that below the critical velocity a thread of colored liquid introduced into the center of a stream of water flowing through the tube remained in the center and sharply defined for the whole tube length, but at the critical velocity it disappeared.

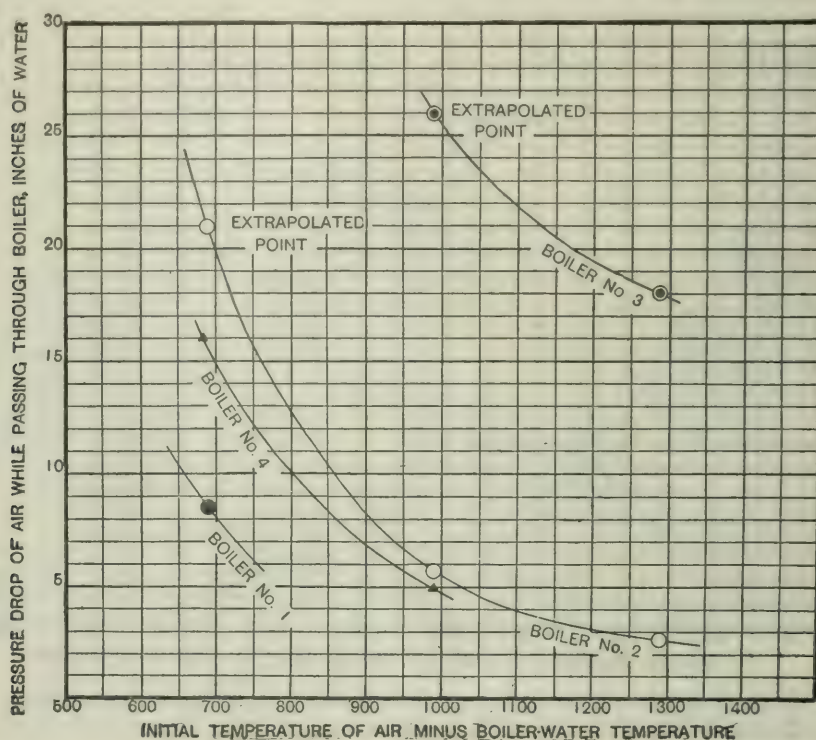


FIGURE 40.—Pressure drops of air while passing through boilers as functions of the initial air minus the boiler-water temperatures for a rate of heat absorption always requiring 2 square feet of heating surface per boiler horsepower. (The points for boiler No. 4 were obtained by extrapolation.)

On November 24, 1904, Prof. Reynolds presented to the British Royal Society a paper on the subject written by Drs. H. T. Barnes and E. G. Coker of McGill University. This excellent paper, entitled "The flow of water through pipes: Experiments on stream-line motion and the measurement of critical velocity" contains the results of many experiments and the conclusions based on them.

Doctors Barnes and Coker conceived the idea of determining the critical velocity of the water in any particular tube by heating the tube along a considerable portion of its length, placing the bulb of a small thermometer in the center of the emerging stream, and gradually opening a valve beyond the thermometer until the beginning of eddying was announced by a sudden rise of the thermometer.

The authors of this bulletin tried similar experiments with a stream of air flowing through a glass tube about 1.5 inches (4 cm.) in diameter. They used exclusively the colored-stream-line method, and introduced into the center of the tube, a little distance inside its entrance, streams of smoke, nitric oxide, and ammonium chloride, all in a vain attempt to get something dark enough to photograph. To the eye the appearance of a critical velocity under certain conditions (temperature of air 15°C ., estimated velocity about 1 meter per second) was unmistakable. The work had to be discontinued because the laboratory was moved.

On referring to figures 6, 14, and 19 one will notice that all the true-boiler-efficiency curves for little multitubular boiler No. 3 have a sharp rise after 100 feet per second velocity; it seems inconceivable that the stream lines of flow remained at all parallel up to this velocity, and yet the sudden increase in heat absorption may possibly have been due to the air velocity reaching a critical value—if there be any such thing as the sudden appearance of a second more violent eddying. Strong evidence against this supposition is afforded by the fact that the upward turns appeared at higher velocities, the higher the initial air temperature.

SUMMARY OF DEDUCTIONS FROM THE LABORATORY EXPERIMENTS.

When the temperature of the air entering a boiler remains constant, the rate of heat absorption by convection by the heating surface of the boiler flues increases very nearly in direct proportion to the velocity of the gas passing over the heating plate. This is particularly true beyond the point of the critical velocity.

When the velocity of the air remains constant, the rate of heat absorption increases as the initial temperature rises, but not in direct proportion to the rise; the increase in the rate of heat absorption becomes smaller for equal rises of initial temperature as the initial temperature becomes higher.

With air velocities greater than the critical velocity the true boiler efficiency is nearly constant for any rate of working and for any initial temperature.

The critical velocity increases when the temperature rises, and seems to drop when the diameter of the flues increases.

Increasing the diameter of the flues decreases their efficiency as heat absorbers—that is, flues of large diameter are less efficient than flues of small diameter of the same length, although the large flues have more heating surface. The higher efficiency of small tubes is due to the fact that the average distance of each particle of the gas from the flue surface is shorter, and therefore each particle of gas comes oftener in contact with the surface. It is not the total area of the heating

surface that determines the efficiency of a boiler, but the cross sectional area of the gas passage and the arrangement of the surface with respect to the flow of gas.

It seems that a large number of small flues having the same total cross-sectional area as a few flues of larger diameter require a smaller pressure drop to push the same weight of gas through than do the large flues.

Increasing the length of flues increases their efficiency, but not in direct proportion; the increase in efficiency becomes smaller with every successive addition to the length of the flue, so that to make flues larger than some definite length would not be good economy.

Most of the resistance to the passage of air through the flues of a tubular boiler is at the entrance to the tubes; adding to the length of the flues increases the resistance but little.

SPECIAL OBSERVATIONS DURING TESTS ON LARGE BOILERS THAT CORROBORATE THE RESULTS OF EXPERIMENTS WITH LABORATORY BOILERS.

During the winter of 1907-8, after the laboratory experiments with small boilers were completed, the authors were detailed to make comparative tests of briquetted and run-of-mine coal on a torpedo boat at the United States navy yard at Norfolk, Va. Later in the same winter they were instructed to make a similar series of tests on a freight locomotive at the yards of the Seaboard Air Line Railway at Portsmouth, Va. During both of these series of tests such observations were taken as would shed light on the extent of the applicability to large boilers of the principles of heat transmission deduced from the laboratory experiments. These observations are presented and discussed in the following pages.

The principle which the results of these observations demonstrate is that without lowering the true boiler efficiency to any extent, the capacity of many boilers can be increased by passing more gases through them.

TESTS WITH THE NORMAND BOILER OF THE TORPEDO BOAT BIDDLE.

The general shape and the principal features of this boiler are shown in figure 41. The boiler was of the water-tube type and consisted of a steam drum and two mud drums, the former being connected to the latter with 1,800 tubes of 1 inch internal diameter. The furnace was under the steam drum between the two nests of tubes and the mud drums. In the rear of the boiler there were two large tubes connecting each mud drum to the steam drum and serving as downcomers for the circulation of the water.

The total heating surface of the boiler was 2,780 square feet; the area of the grate surface was 60 square feet.

Gases were forced through the boiler by keeping the fireroom under pressure, the pressure being maintained by means of a 5-foot pressure fan.^a

^a For a complete description of the boiler and the results of tests, see Bulletin 33, Bureau of Mines, "Comparative tests of briquetted and run-of-mine coal on the torpedo boat *Biddle*," by the present authors.

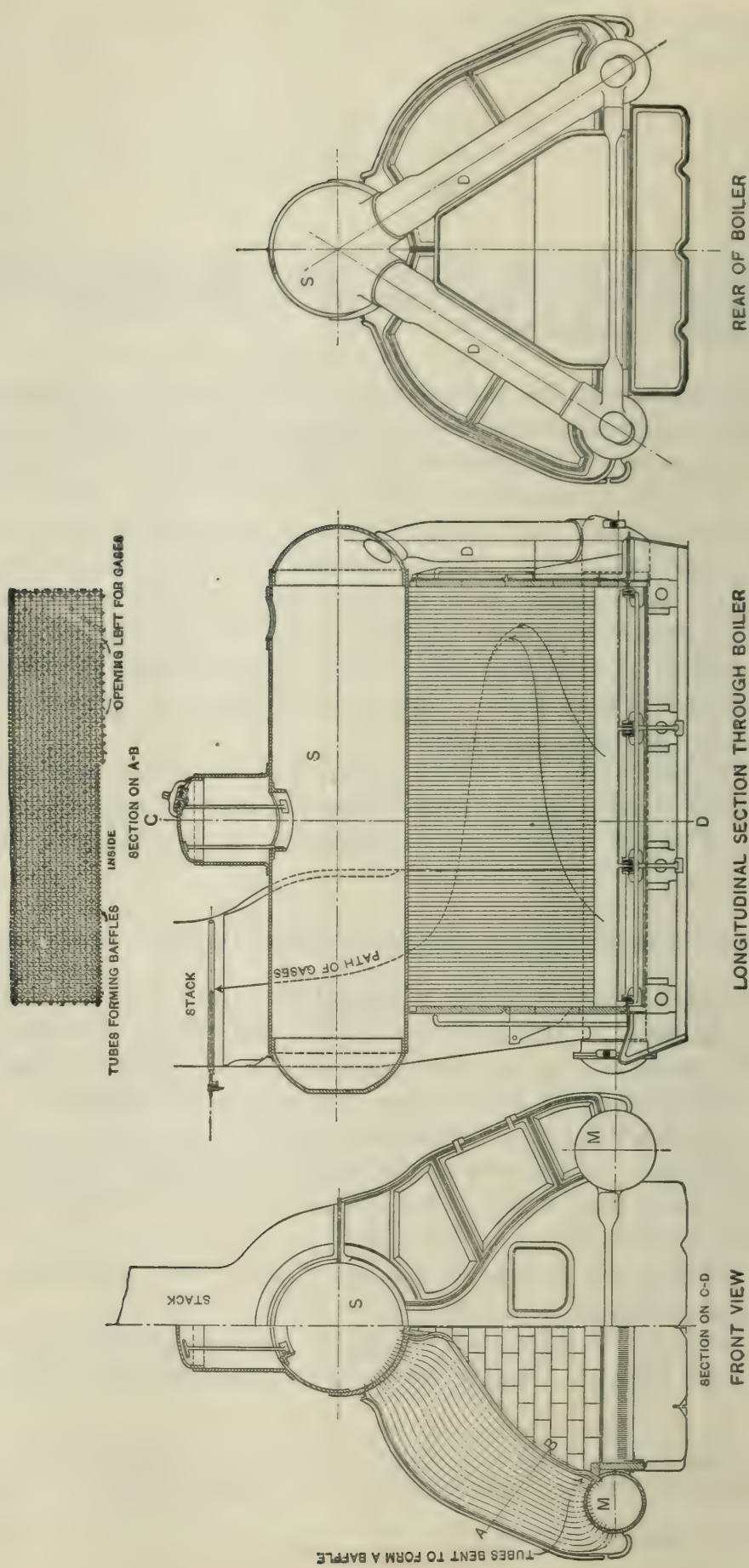


FIGURE 41.—Normand water-tube boiler, in torpedo boat *Biddle*.

TABLE 11.—Data from tests on a Normand boiler and calculations based thereon.^a

Pounds of dry fuel fired per square foot of grate surface per hour.	Temperature of boiler water.	Temperature of gases leaving boiler tubes.	Temperature in furnace, as read by a Wanner optical pyrometer.	Column 4 minus column 2.	Column 4 minus column 3.	Column 6 divided by column 5, giving convection component of the true boiler efficiency.	Boiler horsepower developed.
1	2	3	4	5	6	7	8
	° F.	° F.	° F.	° F.	° F.	° F.	
19.64	380	646	2,367	1,987	1,721	86.6	318.3
20.29	379	597	2,347	1,968	1,750	89.0	339.2
20.41	380	583	2,633	2,253	2,050	90.9	346.5
21.12	381	681	2,460	2,079	1,779	85.6	332.4
21.14	380	583	2,502	2,122	1,919	90.5	335.3
23.95	383	637	2,442	2,060	1,805	87.6	363.8
29.07	383	601	2,498	2,115	1,897	89.6	435.3
29.62	380	739	2,673	2,293	1,934	84.7	460.1
29.98	383	668	2,881	2,498	2,213	88.5	444.5
30.10	380	655	2,273	1,893	1,618	85.4	441.5
31.68	379	696	2,662	2,283	1,966	86.1	490.4
37.28	384	709	2,765	2,381	2,056	86.3	548.1
41.20	380	739	2,615	2,235	1,876	84.0	623.3
45.26	379	797	2,921	2,542	2,124	83.6	671.3
b 45.29	379	919	2,586	2,207	1,667	75.6	596.4
45.41	380	728	2,624	2,244	1,896	84.6	656.6
51.76	382	752	2,950	2,568	2,198	85.5	748.2
64.83	379	807	3,097	2,718	2,290	84.3	850.0
71.48	380	817	3,070	2,690	2,253	83.8	914.4

^a These tests and the calculations based thereon are for the purpose of showing the approximate constancy of the true boiler efficiency in practice. (The actual fuel consisted in different tests of run-of-mine coal, some large "square" briquets and some small nearly "round" ones, both made from part of the run-of-mine coal, but it is unnecessary to denote the distinction here.) No tests are omitted because they "do not fit the theory." Tests are in order of ascending values in column 1.

^b The flue temperature for this test was in all probability much lower than 919° F., as an examination of the original log sheets shows, so that the true boiler efficiency of 75.6 per cent is undoubtedly several per cent low. This efficiency is plotted, but was disregarded in drawing the curve.

In Table 11 are given observations of temperatures in 19 tests made on the Normand boiler of the torpedo boat *Biddle*. These temperatures are used in computing the true boiler efficiency for the purpose of comparing the latter with the rate of combustion. The headings of each column of the table may, perhaps, be supplemented by the following remarks:

In column 1 the word fuel is used to designate either briquets or run-of-mine coal.

In column 2 the temperature of boiler water is obtained from a table of steam temperatures and corresponds to the temperature of saturated steam at the average pressure recorded during each test.

In column 3 the temperature of the gases leaving the heating tubes of the boiler was measured in the uptake about 2 feet above the steam drum, either with a mercury thermometer or a platinum thermocouple. The values given are substantially correct, though a little low.

Column 4, the temperature in the furnace was measured with a Wanner optical pyrometer, which was standardized before each test. The readings were taken through a specially made hole in the back of the furnace.

All the temperatures given in the table represent the average of 15 or 20 readings.

Column 5 is the excess of furnace temperature above the temperature of the water in the boiler; in other words, it is the available temperature drop, which is nearly proportional to the heat available for absorption.

Column 6 is the actual temperature drop of the gases due to the abstraction of heat in passing through the boiler, and is nearly proportional to the heat absorbed by the boiler.

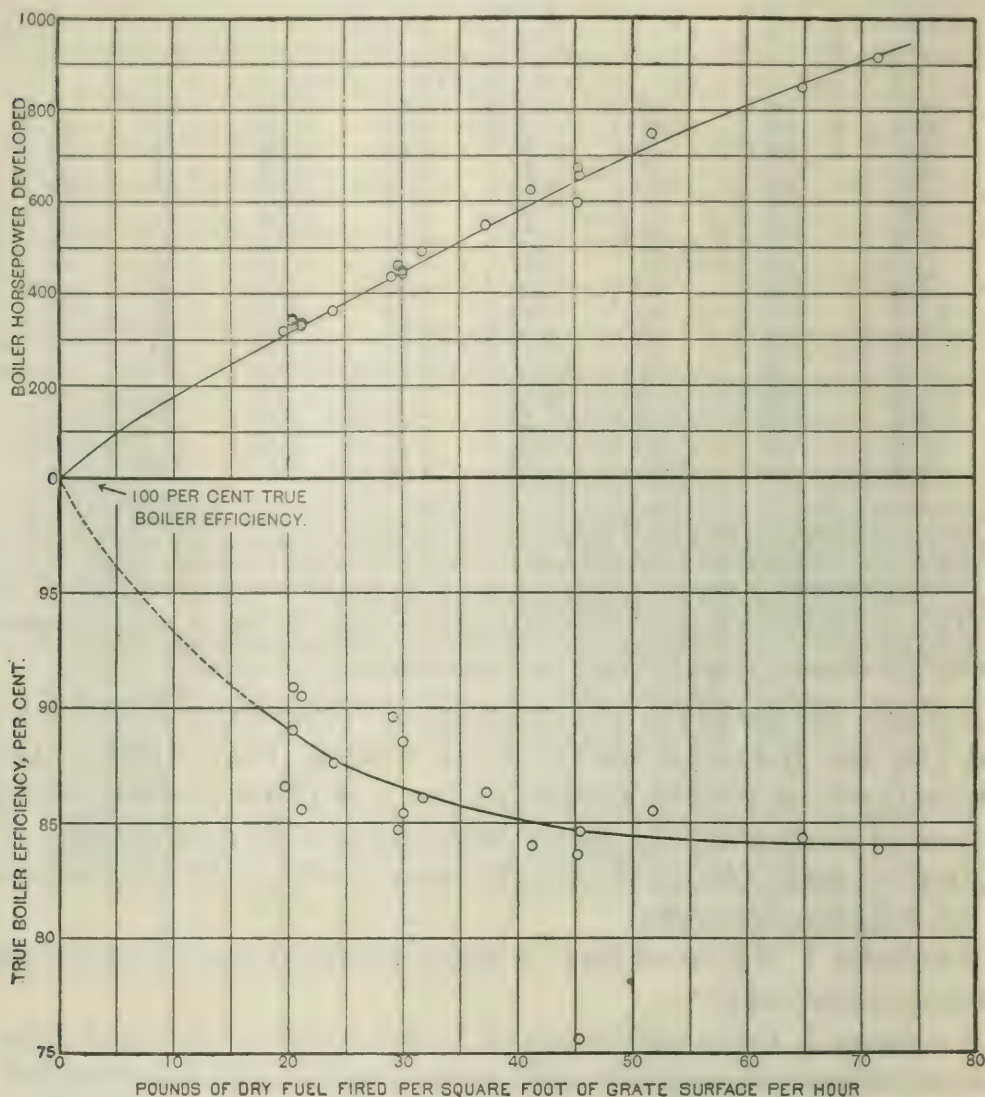


FIGURE 42.—True boiler efficiency and boiler horsepower platted against pounds of dry fuel fired per square foot of grate surface per hour. Results of tests in a Normand water-tube boiler.

Column 7 gives the true boiler efficiency obtained by dividing column 6 by column 5. This computation is the same as is used for the small boiler experiments and is explained on page 31.

Column 8 gives the boiler horsepower which the boiler developed in each test.

Figure 42 has been platted with figures of column 1 as abscissæ, and at the top with figures of column 8 and at the bottom with figures of column 7 as ordinates.

The curve passing through the upper group of points shows the relation between the rate of coal combustion and the rate of heat

absorption. Since approximately the same weight of air was used per pound of coal, the rate of combustion is closely proportional to the weight of gases passing through the boiler, and therefore the curve shows the relation between the weight of gases passing through boiler and the rate of heat absorption.

The curve passing through the lower group of points shows the relation between the true boiler efficiency and the rate of combustion, or the weight of gas passing through the boiler.

Figure 42, then, shows relations for the Normand torpedo-boat boiler similar to those shown by figures 4, 5, 6, and 7 for the small

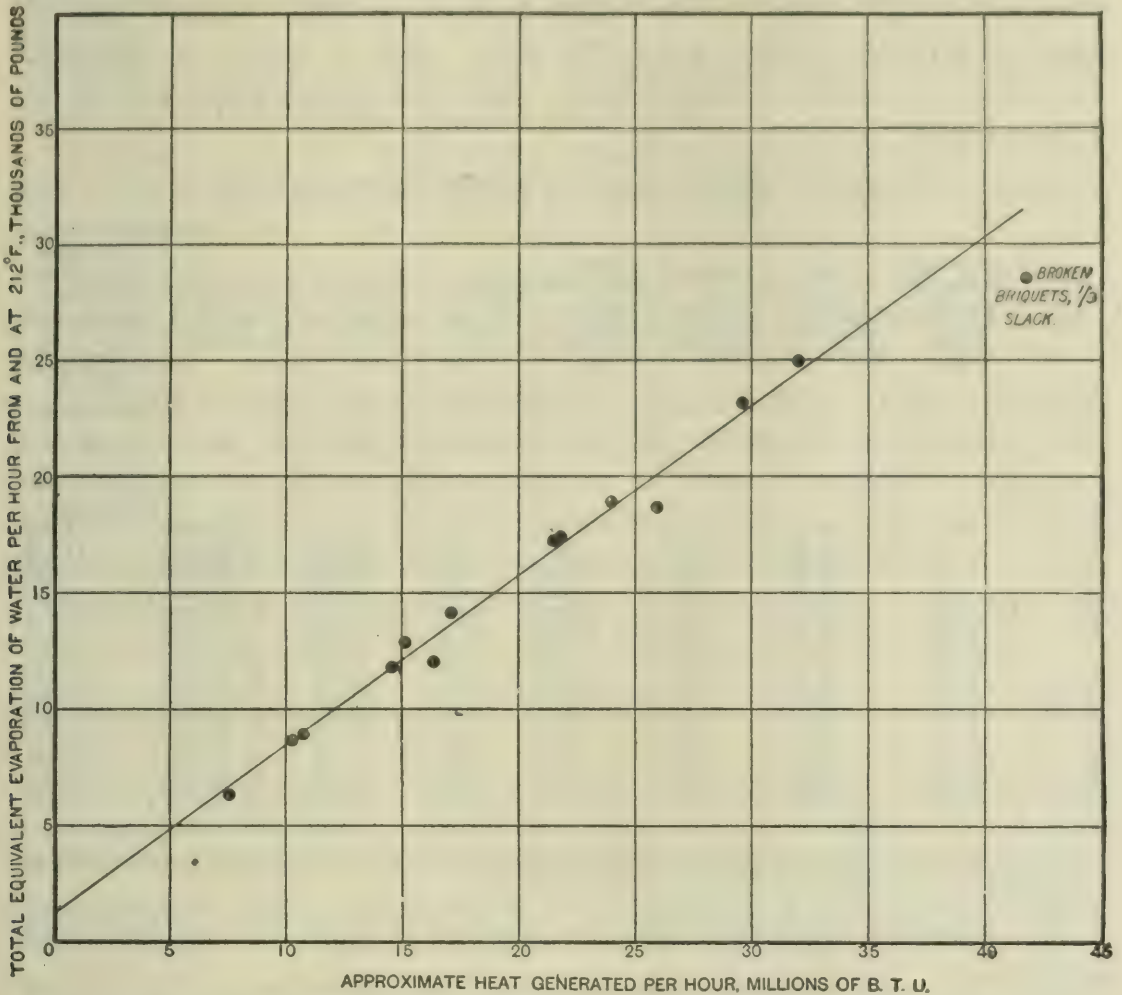


FIGURE 43.—Rate of evaporation as a function of rate of heat evolution in a locomotive boiler.

laboratory boilers. It will be noticed that the curves of the Normand boiler have shapes similar to the curves for the small boilers—that is, the rate of heat absorption increases almost in direct proportion to the increase in weight of the gases passing through the boiler; also the true boiler efficiency remains nearly constant throughout the range of the weight of gas. This boiler develops a boiler horsepower on less than 3 square feet of heating surface, a fact that indicates that the heating plates of the boiler are able to take care of all

the heat that the gases are able to deliver. It may be said that, in general, the conclusions drawn from the experiments with small laboratory boilers hold true for the Normand boiler.

TESTS WITH A LOCOMOTIVE BOILER.

The data given in Table 12 and platted in figure 43 are here presented as a demonstration that in a locomotive boiler the rate of heat absorption is very nearly proportional to the rate of heat generation; that is, the boiler is able to absorb the heat as fast as the latter is delivered to it.

The locomotive on which the tests were made was of a modern 10-wheel freight type. The boiler had in all 2,584 square feet of heating surface. There were 328 flues, each 2 inches in internal diameter and 14 feet 2 inches long. The area of the grate was 29.65 square feet.^a

TABLE 12.—Data from tests on a freight-locomotive boiler.

Test No.	Pounds of combustible, per hour for whole grate. ^a	Heat value of 1 pound of combustible(B. t. u.)	CO loss per pound of combustible.	Unaccounted-for loss per pound of combustible.	Column 4 plus column 5.	Column 3 minus column 6.	Column 2 multiplied by column 7.	Code item 63, equivalent evaporation per hour from and at 212° F.
1	2	3	4	5	6	7	8	9
								Pounds.
1.....	655.4	15,705		21	21	15.684	10,270.000	8,681
2.....	955	15,720	174	307	481	15.239	14,570.000	11,840
3.....	1,026	15,782	79	740	819	15.963	16,380.000	12,023
4.....	494	15,818	186	335	521	15.297	7,570.000	6,312
5.....	729	15,846	107	949	1,056	14.790	10,780.000	8,977
6.....	1,166	15,955	176	1,075	1,251	14.704	17,140.000	14,153
7.....	1,450	15,810	139	648	787	15.023	21,800.000	17,381
8.....	2,074	15,865	113	1,484	1,587	14.278	29,600.000	23,151
9.....	1,960	15,688	117	2,326	2,443	13.245	25,950.000	18,689
10.....	1,042	15,833	162	1,102	1,264	14.569	15,180.000	12,889
11.....	2,155	15,743	20	865	885	14.858	32,000.000	24,941
12.....	1,632	15,915	227	998	1,225	14.690	24,000.000	18,398
13.....	1,533	15,680	38	1,633	1,671	14.009	21,500.000	17,268
14.....	2,822	15,865	163	888	1,051	14.814	41,800.000	28,528

^a This item has been corrected for carbon in ash-pit refuse. The word "combustible" is used loosely as a substitute for the phrase "coal free from moisture and ash."

The following explanation of the columns of Table 12 may be found useful.

Column 2 gives the pounds of "combustible" ascending from the grate per hour. The values given were obtained by subtracting from the weights of coal fired the weights of moisture and ash as determined by chemical analysis, and also the weights of the combustible falling through the grate into the ash pit.

Column 3 gives the heat value of 1 pound of "combustible."

Column 4 gives the heat lost in burning carbon to CO instead of to CO₂. The values are figured from the chemical analyses of the flue gases.

^a For a more detailed description of the locomotive and the tests, see Bulletin 34, Bureau of mines, "Comparative tests of briquetted and run-of-mine coal in a locomotive boiler," by the present authors.

Column 5 is the "unaccounted-for" column in the heat balance. It gives the heat lost by the escape of combustible gases and vapors other than CO, plus the heat lost in sparks, plus the heat radiated. The radiation loss is included because it is embodied in the unaccounted-for item; there is no way to estimate it even approximately.

Column 6 is the sum of columns 4 and 5 and shows the part of the heat value that was put into the furnace but was not delivered to the boiler, and therefore the boiler had no chance to absorb it. In case of the radiation loss part of the heat is lost before it reaches the boiler, but most of it after the boiler has first absorbed it, and therefore it can not justly be charged against the boiler's ability as a heat absorber.

Column 7 shows approximately the heat delivered to the boiler per pound of "combustible" actually consumed.

Column 8 gives the total heat actually evolved in the furnace and delivered to the boiler per hour; in other words, it gives the rate of heat delivery.

Column 9 shows the heat actually absorbed by the boiler per hour, the heat being expressed in terms of pounds of equivalent water evaporated. This column shows the rate of heat absorption.

Columns 8 and 9 have been platted in figure 43 with the values of the former as abscissæ, and those of the latter as ordinates. The resulting curve shows that the rate of heat absorption increases almost directly as the rate of heat delivery; that is, when the rate of delivery is doubled, the heat absorbed is nearly doubled.

Figure 43 shows such relation for the locomotive boiler as the upper curves of figures 4, 5, 6, and 7 show for the small boilers.

GENERAL CONCLUSIONS DRAWN FROM TESTS MADE ON LARGE BOILERS.

The general conclusion that can be drawn from the above-mentioned tests on the Normand water-tube boiler and the locomotive boiler is that the principle of the rate of heat absorption, developed by the experiments with small boilers, holds good to a great extent with large boilers. This principle states that the quantity of heat absorbed by the boiler increases nearly at the same rate as the quantity of available heat supplied to the boiler. In other words, the true boiler efficiency is nearly constant. To double the capacity of a boiler, double the heat generated in the furnace.

If the furnace temperature is always about the same, to double the capacity of a boiler burn about twice the weight of coal and pass twice the weight of gases over the heating plates; to triple the capacity pass three times the weight of gas over the heating plates, and so on. For most boilers, if the combustion of coal is fairly complete, the over-all efficiency as well as the true boiler efficiency will remain nearly constant with almost any rate of combustion.

In practice, when the capacity of a steam-generating apparatus is increased the combustion of the fuel in the furnace is less complete and the over-all efficiency drops. The blame for this drop in over-all efficiency should fall on the furnace and not on the boiler; the boiler generally does its part of the work. Nevertheless, it is not always safe to assume that a boiler will maintain its true efficiency on increasing the rate of heat absorption, because many commercial boilers are probably working on the first part of their true boiler efficiency curves, where the curves are still quite steep.

PART III.

ABSTRACTS FROM WORKS OF OTHER INVESTIGATORS.

SOME PREVIOUS EXPERIMENTAL WORK ON HEAT TRANSMISSION.

For many years the experiments of Peclet on the conductivity of metals were widely accepted; he separated a mass of water of one temperature from a mass of another temperature by a metal plate, stirred both masses more or less, and measured the rate of heat transfer from the hot to the cold water. No one blames Peclet, for his results were far better than any one else had obtained; the blame for their continued acceptance rests on the engineers, and even on the physicists who thought no deeper for years after the kinetic theory of gases and liquids had become well known. Probably Prof. William Thompson (Lord Kelvin) was among the first to find the trouble. In the first edition of the *Encyclopædia Britannica*, in his article on "Heat," he quotes Peclet's best results and says that the trouble is to get the heat from water to metal and from metal to water; he remarks that the only question to be answered concerning Peclet's values for the heat conductivity of metals is whether his values are tens or hundreds of times too small. The reader is referred to any recent book on physics for a discussion of recently devised methods for obtaining the true heat conductivities of metals. The conductivities found are surprisingly high.

DEDUCTIONS OF REYNOLDS.

In 1874, many years after the publication of Peclet's experiments, Prof. Osborne Reynolds presented before the Literary and Philosophical Society of Manchester, England, a paper entitled "On the extent and action of the heating surface for steam boilers," in which he approached the problem of how heat is imparted to the heating sheets of a boiler by convection. Prof. Reynolds especially mentioned the influence of the velocity of the gas on the rate of heat transfer and also the immense value of the principle if intelligently applied in steam-boiler practice; nevertheless the paper was completely overlooked by engineers until the attention of Prof. John Perry was accidentally directed to it a few years ago. If a less able engineer than Prof. Perry had examined this paper, it would still be slumbering. Although Perry emphasized the practical importance of the principles evolved in Prof. Reynolds's paper and pointed out many

commercial applications, he left his work so mathematically involved that it is about as good as sealed. Although his book, entitled "The Steam Engine and Gas and Oil Engines," has been out a number of years (published in 1900), and other parts of the book are widely quoted, the part on the transfer of heat through boiler plates has received very little attention.

Reynolds's paper is so clear and excellent that part of it is quoted below:

The heat carried off by air or any fluid from a surface, apart from the effect of radiation, is proportional to the internal diffusion of the fluid at and near the surface; that is, is proportional to the rate at which particles or molecules pass backward and forward from the surface to any given depth within the fluid; thus, if AB be the surface and ab an ideal line in the fluid parallel to AB , then the heat carried off from the surface in a given time will be proportional to the number of molecules which in that time pass from ab to AB ; that is, for a given difference of temperature between the fluid and the surface. This assumption is based on the molecular theory of fluids.

Now the rate of this diffusion has been shown from various considerations to depend on two things:

1. The natural internal diffusion of the fluid when at rest. 2. The eddies caused by visible motion which mix the fluid up and continually bring fresh particles into contact with the surface.

The first of these causes is independent of the velocity of the fluid, and if it be a gas is independent of its density, so that it may be said to depend only on the nature of the fluid.

The second cause, the effect of eddies, arises entirely from the motion of the fluid and is proportional both to the density of the fluid, if gas, and the velocity with which it flows past the surface.

The combined effect of these two causes may be expressed in a formula as follows:

$$(1) \quad H = A\theta + Bwv\theta,$$

where θ is the difference of temperature between the surface and the fluid, w is the density of the fluid, v its velocity, and A and B constants, depending on the nature of the fluid, H being the heat transmitted per unit of the surface in a unit of time.

If, therefore, a fluid were forced along a fixed length of pipe which was maintained at a uniform temperature greater or less than the initial temperature of the gas, we should expect the following results:

1. Starting with a velocity zero, the gas would then acquire the same temperature as the tube.

2. As the velocity increased the temperature at which the gas would emerge would gradually diminish, rapidly at first but in a decreasing ratio until it would become sensibly constant and independent of the velocity. The velocity, after which the temperature of the emerging gas would be sensibly constant, can only be found for each particular gas by experiment; but it would seem reasonable to suppose that it would be the same as that at which the resistance offered by friction to the motion of the fluid would be sensibly proportional to the square of the velocity; it having been found both theoretically and by experiment that this resistance is connected with the diffusion of the gas by a formula—

$$(2) \quad R = A'v + B'wv^2;$$

and various considerations lead to the supposition that A and B in (1) are proportional to A' and B' in (2). The value of v which this gives is very small, and hence it follows that for considerable velocities the gas should emerge from the tube at a nearly constant temperature whatever may be its velocity.

This, as I am about to point out, is in accordance with what has been observed in tubular boilers as well as in more definite experiments.

In the locomotive the length of the boiler is limited by the length of the tube necessary to cool the air from the fire down to a certain temperature, say 500° F.

Now, there does not seem to be any general rule in practice for determining this length, the length varying from 16 feet to as little as 6 feet, but whatever the proportions may be, each engine furnishes a means of comparing the efficiency of the tubes for high and low velocities of the air through them. It has been a matter of surprise how completely the steam-producing power of a boiler appears to rise with the strength of blast or the work required from it, and as the boilers are as economical when working with a high blast as with low, the air going up the chimney can not have a much higher temperature in the one case than in the other. That it should be somewhat higher is strictly in accordance with the theory as stated above.

It must, however, be noticed that the foregoing conclusion is based on the assumption that the surface of the tube is kept at the same constant temperature, a condition which it is easy to see can hardly be fulfilled in practice.

WORK OF PERRY.

It was from this paper by Prof. Reynolds that Prof. Perry took his cue and later developed the theory set forth in his before-mentioned book on "The Steam Engine and Gas and Oil Engines." The following is a brief abstract of Prof. Perry's work:

When fluid is in motion filling a pipe, we know that there is a thin film or layer of fluid entangled among the molecules of the solid surface which is at rest, that is, it has no average velocity relatively to the solid. Let us consider how heat gets into this film from the moving fluid. It is difficult to say whether one ought or ought not to take entrance of heat to this layer of motionless fluid as entrance to the metal itself. There is equalization of the momentum, and there may be equalization of the temperature.

Now, suppose n molecules per second to enter this layer and the same number to leave it; each of them enters with an average kinetic energy proportional to T , the average temperature (absolute) in the pipe, and leaves with t_1 the temperature of the layer, and an average momentum in the axial direction proportional to v if v is the average axial velocity in the pipe.

There is a want of exactness in my definition of these averages which is, I think, the only weakness in this investigation. Now, axially directed momentum given to the layer per second is what we mean by force of friction F .

So that per unit area,

$$(3) \quad F \propto nv,$$

and the heat H or kinetic energy per second per unit area

$$(4) \quad H \propto n(T - t_1).$$

Hence,

$$(5) \quad H \propto F(T - t_1) \div v.$$

Of course, when v is 0 we can not use (3) in (4) to find (5), but we shall only use our equations in cases where v has some value.

In the standard books on friction of fluids in pipes, the law is given

$$(6) \quad F \propto wv^2$$

where w is the weight of the fluid per unit volume, and v is the average axial velocity in the channel. I am informed by Prof. O. Reynolds that the results of his 1883

paper in the Philosophical Transactions are applicable to gases, and taking his index there as 2 we have the same formula as (6); (5) and (6) give us

$$(7) \quad H = C' w v (T - t_1)$$

where C' is a constant.

Perry's equation (7) is the same as Osborne Reynolds's equation (1) if the first term on the right side of the equation is made to equal zero. It is therefore more correctly applicable only to cases where the motion of the gas exceeds the critical velocity.

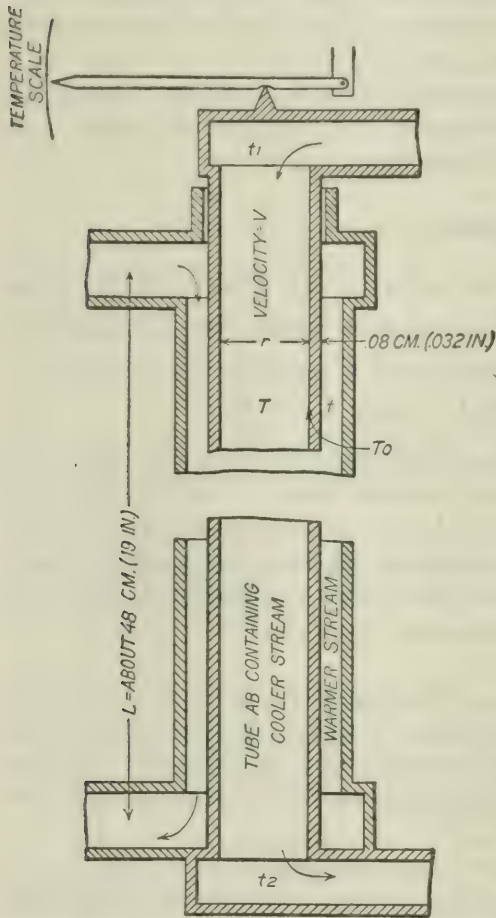


FIGURE 44.—Diagrammatic representation of Stanton's apparatus.

EXPERIMENTS BY STANTON.

In the Philosophical Transactions of the Royal Society of London, volume 190 (year 1897), pages 67 to 88, is an excellent paper by T. E. Stanton, entitled "Passage of heat between metal surfaces and liquids in contact with them."

The paper is based on the results of experiments with two streams of water at different temperatures. These two streams were forced at varying velocities through two concentric tubes, one surrounding the other. The results showed that at quite different velocities the change in temperature produced by a stream while passing a given length of the tubing was very much the same; that is, twice as much heat passed through the inner tube when the velocity of the flowing streams was doubled.

Stanton's work was so well done, and the experimental results so thoroughly studied and presented, that it deserves the widest publicity; nevertheless it is almost unknown in the United States. Incorporated with the data is some excellent mathematical work which, however, has perhaps caused many readers to turn away. The authors of this bulletin are not especially fond of mathematical theory, but the treatment in Stanton's paper is so admirable that they have decided to transcribe some of the paper, annotated throughout. They would not have done this except that the transactions of the Royal Society are in only a few of the large libraries of this country.

The apparatus used by Stanton consisted essentially of two metal tubes placed vertically, one inside the other, with a thin annular

space between, as indicated in figure 44. The matter following is either paraphrased or quoted, and all passages occurring within brackets are inserted as explanations by the authors of this bulletin, and must be taken solely as the latter authors' opinions, not in any degree binding on Mr. Stanton or others:

When the quantities of water flowing through tube AB and through the annular space per second were equal, the temperature of the metal of tube AB was the same at all points along its length; the cooler water flowed down the pipe AB, and the warmer water down the annular space around AB.

The temperature of the pipe at any cross section will not necessarily be the mean between T and t [which are the temperatures of the water inside and outside of AB at any point under discussion] at that section, but if the total temperature fall be small (6° C. or less), then under the following conditions of flow we may fairly assume that the ratio of the difference of temperature between (jacket water and wall) and between (wall and water in AB) is constant for the whole length of pipe, and hence that the temperature of the pipe is the same throughout its length. ["The condition of flow" is that the velocities of both streams of water be above their critical velocities, that is, above those velocities at which parallel stream lines give way to continuous and violent eddying.]

DESCRIPTION OF APPARATUS.

Three sizes of internal pipe (AB) were used, of drawn copper, each 48 cm. [19 inches] long, of 0.08 cm. [0.032 inch] thickness of walls, and of internal diameters of 1.39, 1.07, and 0.736 cm. [0.55, 0.42, and 0.29 inch]. The [outside] jacket pipes were of brass, the widths of the jacket spaces being, respectively, 0.165, 0.65, and 0.16 cm. [0.065, 0.26, and 0.063 inch].

The temperature of the copper pipes was accurately obtained by measuring their change of length, this change being shown on a scale by a multiplying lever.

All parts which it was essential to keep warm or cool were carefully lagged with cotton wool.

RESULTS OF EXPERIMENTS.

Let

T_0 =the temperature of the inner surface of the [inside] pipe in degrees centigrade,

t =the mean temperature of the water in the [inside] pipe at any cross section,

v =the velocity of the water through the [inside] pipe in centimeters per second,

p =the pressure of the water [apparently in centimeters of mercury],

r =the [inside] radius of the [inside] pipe in centimeters,

and L =the length of the pipe in centimeters.

"Then, for a small element of the internal surface of the [inside] pipe $=2\pi r dl$, we may write, for the case of transmission of heat from metal to water, Heat transmitted [per second] $=dH=K \cdot 2\pi r dl \cdot \phi(T_0, t, (T_0-t) \cdot p, v, r)$, where K and the function ϕ are to be determined by experiment.

"Each variable in the above equation was studied separately. To determine the effect of v (velocity of water), series of experiments were made at various velocities, in which values of T_0 , t , (T_0-t) , p , and r remained constant.

As to variations of pressure of water only (p), between 1 and 2 atmospheres, the rate of heat transmission was "practically independent of pressure."

As to variations in velocity, "the heat transmitted for the given range of temperature is nearly proportional to the velocity of the water, even when (T_0-t) varies between 5° and 40° C." [(T_0-t) is the difference in temperature between the inner surface of the metal pipe AB and the water flowing in its interior.]

As to the effect of varying the ranges of temperature: "Now, if the heat transmitted under these conditions varied simply in [with] the range of temperature, we should have had for the whole surface of the pipe

$$C(T_0 - t)ds = Wdt$$

$$\text{or } C_1 S = W \log \frac{T_0 - t_1}{T_0 - t_2}$$

where S = the whole surface of the pipe,

C_1 = a constant,

and W = weight of water flowing through the pipe per second.

"Now, the values of $\log \frac{T_0 - t_1}{T_0 - t_2}$ for the experiments quoted are 0.222, 0.218, and 0.213, respectively, which show that the heat transmitted is proportional to $(T_0 - t)$ multiplied by a function of the temperature T_0 ." [That is, it appears that the rate of impartation of heat from the copper to the cooler water at any point where the temperature of the inside surface of the metal is T_0 and that of the water t , is not directly proportional to the temperature difference between the metal and the water $(T_0 - t)$, but increases a little with increasing temperature of the metal, all other things equal.]

As to the effect of varying the initial temperature of the water, t , flowing down the inner pipe, while keeping constant the temperature difference between the inner surface of the copper pipe and the entering water in it $(T_0 - t_1)$, and also while keeping constant the velocity of the water, v , and the radius of the pipe, r , results under these conditions show a considerably higher rate of heat transmission at higher initial temperatures [care being taken to raise the temperatures of the inner surface of the copper tube and the water an equal number of degrees, so as to keep constant the temperature difference between the two $(T_0 - t_1)$].

For a pipe of given diameter experiments seem to show that the rate of heat transmission from the pipe to water flowing through it is given by an expression of the form

$$(8) \quad dH = K \cdot 2\pi r \cdot dl \cdot (T_0 - t) \cdot f(v) \cdot F(T_0) \cdot \Phi(t)$$

[Where dH = an extremely small quantity of heat transmitted in an extremely short length of the inner pipe in a unit of time;

$2\pi r dl$ = the internal area of an extremely short length of a tube l cm. long (l being of course a very small fraction of 1 centimeter) and r cm. in internal diameter;

$(T_0 - t)$ = the temperature difference between the inner surface of the metal tube at the cross section under discussion and the water at that section (this section being anywhere along the length of the tube);

$f(v)$ = a function of the velocity of the water in the pipe;

$F(T_0)$ = a function of the temperature of the inside of the tube at that cross section (supposed, moreover, to be the same all along the tube);

and $\Phi(t)$ = a function of the temperature of the stream of water at the cross section under discussion].

It is also seen that the values of $F(T_0)$ and $\Phi(t)$ do not vary much from unity and may probably be put in the forms

$$F(T_0) = 1 + \alpha T_0 \text{ and } \Phi(t) = 1 + \beta t,$$

where α and β are constants to be determined by experiment. [That is, it is probable that the increase in the rate of heat transmission resulting from a higher inside temperature of the pipe (T_0) , is increased (or diminished) some certain small amount for each additional degree centigrade; and it is probable, too, that the rate is also increased (or diminished) some other certain small amount for each additional degree of temperature of the water at that cross section. α and β are very small decimals of constant value.]

Again, it is seen that the heat transmitted is nearly proportional to the velocity of the water, perhaps expressed by some equation of the form

$$f(v)=V^n,$$

where n =a number a little less than unity, to be determined by experiments at varying velocity. [That is, it is likely that the increase in the rate of heat transmission does not quite keep up with an increase in the velocity of water flowing through the pipe, all other things being equal.]

In equation 8 the value of K may depend on the diameter of the pipe and the nature of its surface.

Experiments made on the three copper pipes 1.39, 1.07, and 0.736 cm. in diameter did not indicate what the relation was between H and r , except that variation in pipe diameter [twice r] had little effect.

It is shown in the theory that the heat transmission is proportional to the value of r^{n-2} , where $n=1.84$ approximately. [In words, the rate of heat transmission is not quite doubled by doubling the diameter of the tube, nor tripled by tripling the diameter, etc.] This would make the heat transmission across unit area of the surface of the smallest pipe (diameter 0.736 cm. in diameter) about 10 per cent greater than across unit area of the largest pipe (diameter 1.39 cm.) under the same conditions of flow and temperature.

GENERAL THEORY.

According to this theory [of Prof. Osborne Reynolds] the motion of heat from the surface of the pipe follows the same laws as the motion of momentum to the surface, whether by conduction or convection (though not by radiation and absorption through the material, which unquestionably plays an important part in the so-called conduction of water).

Take x as the direction of motion of the water.

Let r =the radius of the pipe,

t =the temperature of the water [at any cross section],

T_0 =the temperature [of the inside surface] of the pipe,

w =the weight of unit volume of water,

p =the pressure of water per unit area,

W =the weight of water discharged per second,

v =the velocity of the water through the pipe,

$P=(1+0.0336t+0.000221t^2)^{-1}$,

and A , B , and n be constants depending on the nature of the surface.

[According to Prof. Osborne Reynolds's results and deductions (at least for small or medium-sized tubes), the critical velocity is

$$V_{\text{critical}} = \frac{P}{84.7\delta} \text{ feet per second,}$$

Where δ =diameter of tube in feet,

and P is a number getting smaller with rising temperature according to the formula

$$P=(1+0.0336t+0.000221t^2)^{-1}.$$

It is evident that the warmer the water the lower the velocity at which eddying begins. Temperatures are all in degrees centigrade. This value for P was experimentally obtained by Prof. Reynolds in some researches previously made.]

Then above the critical velocity the loss of pressure is given by the equation:

$$\frac{dp}{dx} = \frac{P^{2-n}}{(2r)^{3-n}} v^n \cdot \frac{B^n}{A}$$

Writing this in the form

$$(9) \quad \pi r^2 \frac{dp}{dx} = \pi r^2 g W \frac{P^{2-n}}{(2r)^{3-n}} v^{n-1} \frac{B^n}{A} \left(\frac{W}{g} v \right)$$

Now, in (9) $\pi r^2 \left(\frac{dp}{dx} \right)$ is the loss of momentum due to diffusion and convection, so that, according to the above theory, substituting

$$W \frac{dt}{dx} \text{ for } \pi r^2 \frac{dp}{dx} \text{ and } W(T_0 - t) \text{ for } \frac{W}{g} v,$$

the equation for the passage of heat will be

$$(10) \quad W \frac{dt}{dx} = \pi r^2 g \frac{P^{2-n}}{(2r)^{3-n}} v^{n-1} \cdot \frac{B^n}{A} \cdot (T_0 - t)$$

or, writing $W = wv\pi r^2$, the slope of temperature along [down] the pipe is given by

$$(11) \quad \frac{dt}{dx} = \frac{B^n}{A} \cdot \frac{g}{w} \cdot \frac{P^{2-n}}{(2r)^{3-n}} v^{n-2} (T_0 - t)$$

This is on the supposition that the conductivity of the water, as compared with the viscosity, does not enter; but as it probably does, for ultimately it is by conductivity that the heat passes from the walls of the pipe to the water, there will probably be a coefficient $f_P^{(c)}$, the form of which can be determined by experiment.

APPLICATION OF PROF. REYNOLDS'S THEORY TO THE EXPERIMENTS.

Assuming the variation in the value of t to be small, say not greater than 6°C. in the whole length of the pipe, then, integrating equation (11),

$$(12) \quad \log \frac{T_0 - t_1}{T_0 - t_2} = \frac{B^n}{A} \cdot \frac{g}{d} \cdot \frac{P^{2-n}}{(2r)^{3-n}} \cdot v^{n-2} L$$

where t_1 = temperature, in $^\circ\text{C.}$, of water entering the [inside] pipe,

t_2 = temperature, in $^\circ\text{C.}$, of water leaving the [inside] pipe,

L = the length of the pipe in centimeters,

and P^{2-n} = the mean value of P^{2-n} for the water all along the inside of the pipe.

Now, from equation (12) the value of n can be determined by a set of experiments in which P^{n-2} has the same value in each. This was done by plating the logarithmic homologues of $\log \frac{T_0 - t_1}{T_0 - t_2}$ and v , when it was found that the points platted lay all approximately on a straight line, there being no systematic deviation.

For the three pipes used in the experiments the slopes of the logarithmic homologues were found to be:

For pipe I, 1.390 cm. diameter, slope = -0.140 , or $n = 1.86$.

For pipe II, 1.070 cm. diameter, slope = -0.175 , or $n = 1.825$.

For pipe III, 0.736 cm. diameter, slope = -0.170 , or $n = 1.83$.

In pipe I the velocities of the water had values between 28.7 and 123.2 cm. per second.

In pipe III the velocities of the water had values between 60 and 393.7 cm. per second.

It will be seen that the values of n given above correspond with the values we should expect to find for smooth copper pipes from the law of resistances, the value for glass being about 1.73, and for smooth metal rather higher, rising to 2 for rough metal surfaces.

EFFECT OF VISCOSITY AND CONDUCTIVITY.

If the conductivity of the water at the bounding surface be neglected, then for experiments at constant velocity equation (12) gives the value of

$$\frac{\log \frac{T_0 - t_1}{T_0 - t_2}}{P^{2-n}}$$

constant for different values of t .

On reference to the results given in the tables [not printed in this bulletin], it is seen that the value of this expression rises with the value of the mean temperature (t_m) of the water, which seems to show that the conductivity of the water at the boundary has an effect.

It was also seen, in the experiments quoted above, that the heat transmitted depended on the values of T_0 and t , and that this effect would be represented by coefficients of the form

$$(1 + \alpha T_0) \text{ and } (1 + \beta t).$$

[That is, roughly speaking, the rate of heat transmission increases some certain small amount for each additional degree of temperature of the inside surface of the copper tube, and also, independently, some other certain small amount for each additional degree of temperature of the water inside the inner tube. These slight increases are over and above the increases due to increasing the temperature differences ($T_0 - t$) between the metal and the water. At higher temperatures water is "thinner" in the sense of having lower viscosity, in which fact lies this explanation.]

From experiments at constant values of v and t_m , the value of α is found to be $\alpha = 0.004$, and that of β is $\beta = 0.01$. The slope of the temperature of the water in the pipe is then given by

$$(13) \quad \frac{dt}{dx} = \frac{B^n}{A} \cdot \frac{g}{w} \cdot \frac{P^{2-n}}{(2r)^{3-n}} v^{n-2} \cdot (T_0 - t) \cdot (1 + \alpha T_0) \cdot (1 + \beta t)$$

[The physical meaning of this equation may be rather loosely explained as follows: The rise in temperature of the water while passing down any extremely short portion of the pipe is proportional to several factors: First to the constants B , n , and A , which are dependent on the nature and condition of the internal surface of the pipe, and for any one pipe remain the same (unless the pipe be cleaned or polished, for instance); second, to the value of gravity (constant) and inversely proportional to the weight of a cubic centimeter of the water at the cross section under discussion, a weight that is nearly the same all along the pipe; third, to a small extent on the magnitude of the factor P (previously described), and inversely proportional to the diameter of the pipe raised to a little over the first power, $(2r)^{3-n}$; fourth, to the velocity of the water raised to a very small negative power; that is, the velocity of the water affects very slightly the temperature rise per unit length of pipe, the rise decreasing a little with increasing velocities; fifth, to the temperature difference between the inner surface of the pipe and the water ($T_0 - t$); sixth, to the temperature of the inside surface of the pipe to a very slight extent, being increased about four-tenths of 1 per cent for each degree centigrade; seventh, to the temperature of the water to a very slight extent, being increased about 1 per cent for each degree centigrade.]

Now, for smooth metal pipes the value of B^n may be assumed as nearly constant, so for a pipe of given length and diameter, in which the surface temperature is constant,

$$(14) \quad kL = \frac{(2r)^{3-n} v^{2-n} \log \frac{T_0 - t_1}{T_0 - t_2}}{P^{2-n} (1 + \alpha T_0) (1 + \beta t_m)}$$

where k is a constant depending on the nature of the surface of the pipe.

[The commentators see no way to state clearly the physical meaning of this equation; it is used to determine the value of k , which is a factor that engineers desire to be large, for the rate of heat impartation is directly proportional to the magnitude of k . As will be shown, k is larger when the flow of heat is from metal to water than when it is from water to metal.]

For the ranges of temperature obtained in these experiments no sensible error is introduced by taking the mean values of P^{2-n} and t for the experiment and substituting them in equation (14). When the variation in t is considerable, equation (13) must be integrated more exactly.

Applying equation (14) to the results of the experiments on the three copper pipes, the values of k are found to be:

Pipe.	Diameter.	Length.	Value of k .			Number of experiments.
			Maximum.	Minimum.	Mean.	
I.....	cm. 1.39	cm. 47.0	0.0108	0.0104	0.0106	22
II.....	1.07	44.5	.0104	.0100	.0102	13
III.....	.736	46.0	.0103	.0099	.0100	16

If W = the weight of water flowing through the pipe in grams per second, equation (13) may be written

$$(15) \quad W \frac{dt}{dx} = k \pi r^2 \frac{P^{2-n}}{(2r)^{3-n}} v^{n-1} (T_0 - t) (1 + \alpha T_0) \cdot (1 + \beta t)$$

which gives for the transmission of heat from metal to water per square centimeter of the surface of the pipe

$$(16) \quad H = \frac{k}{4} \cdot \frac{P^{2-n}}{(2r)^{2-n}} (T_0 - t) (1 + \alpha T_0) \cdot (1 + \beta t) v^{n-1}$$

H = number of gram-degrees [small calories] per second.

[This equation is one of the goals: It states specifically the amount of heat that any small length of the copper pipe will transmit to cooler water flowing through it, providing the velocity of the water is sufficiently high to keep up constant eddying; for an ordinary clean copper surface n will equal about 1.835, as found herein. For rough surfaces and for other metals it will not be widely different. Inasmuch as the values of all other letters are known in any one case, the number of calories imparted per second may be calculated for any short length of tube, say, 1 cm. The value of H for the whole length of the tube is obtainable preferably by integration, but the result can be approximately attained by a step-by-step process, which will not be gone into here.]

CASE OF TRANSMISSION OF HEAT FROM WATER TO METAL.

"The theory for this case is the same as in the transmission of heat from the surface to the water, so far as convection of the heat is concerned. It seemed probable also that the conductivity coefficients would be the same, but on experiment they were found to differ, the viscosity in this case having a much greater effect. The result of the experiments was that the heat transmitted was nearly inversely proportional to the mean viscosity of the film of liquid at the surface, so that the conductivity coefficient can be put in the form $\frac{K}{P_m}$, in which P_m is the mean value of P for the experiment, and

$$P = \left\{ 1 + 0.0336 \left(\frac{T_0 + t}{2} \right) + 0.000221 \left(\frac{T_0 + t}{2} \right)^2 \right\}^{-1}$$

[This value for P is to be used precisely the same as the previous one explained on page 95; the important distinction is that the simpler one is to be used when the metal is warmer than the water, and this one when the water is warmer than the metal.]

For this case equation (13) now takes the form

$$(17) \quad \frac{dt}{dx} = -\frac{B^n}{A} \cdot \frac{g}{w} \cdot \frac{P^{2-n}}{(2r)^{3-n}} \cdot \frac{v^{n-2}(t - T_0)}{P_m}$$

"The results of experiments made under these conditions are given in Table V [omitted from this bulletin], and the values of k are calculated. These values do not show quite such a high degree of consistency as the values of k deduced from the other experiments, but this is probably due to the difficulty of working under the given conditions. Other experiments in which the heat flowed from water to metal were made, but always with the result that the heat transmitted was sensibly inversely proportional to the mean viscosity of the film at the surface, the deviation never being greater than 7 per cent."

[In Table V the values of k were 0.00743, 0.00726, 0.00778, 0.00763, and 0.00765, averaging 0.00755, which is only three-quarters as large as when the flow of heat is in the other direction; that is, the flow of heat from water to metal is only 75 per cent as rapid as from metal to water, all other conditions being the same; of course this is known to be true only within the range of these experiments.]

WORK OF NICOLSON.

In January, 1909, John T. Nicolson, professor of mechanical engineering at the University of Manchester, delivered a lecture on heat transmission before the Junior Institute of Engineers. In this lecture he discussed the results of heat-transmission experiments made on a modified Cornish boiler and on small laboratory apparatus. A full account of his lecture is given in *Engineering* of February 5 and 12, 1909. Since the results of Prof. Nicolson's experiments and his deductions therefrom contribute much toward establishing the laws of heat transmission into steam boilers, a brief statement of the substance of his lecture is given in the following paragraphs.

In his lecture Prof. Nicolson discussed mainly two sets of experiments, one on a large boiler, and one on a small laboratory apparatus.

The large boiler used was an ordinary Cornish boiler 6 feet 6 inches in diameter, and 26 feet long. It had a furnace flue 41 inches in diameter, consisting, as usual, of short lengths jointed with Adamson rings. The first 6 feet of the flue from the front end was occupied by an ordinary grate having an area of 19 square feet; the second portion, about 8 feet long, back of the bridge wall, was lined with fire brick, and formed a combustion chamber; the third portion, about 10 feet long, was nearly filled with a cylindrical vessel made of boiler plate that measured 39 inches in outside diameter and contained water. The front end of this vessel was closed by a domed head. Thus an annular space about 1 inch wide, 40 inches in mean diameter and 10 feet long was formed between the inside surface of the flue and the outside of the water vessel. Through this annular space all the products of combustion were forced.

When about 1,200 pounds of coal were burned per hour, the products of combustion were cooled from a temperature of about 2,300° F. in the combustion chamber to 800° F. at the outlet from the annular space. At this rate of combustion the quantity of heat transmitted per square foot of heating surface per hour was about 34,450 B. t. u., which gave 1 boiler horsepower on less than 1 square

foot of heating surface. The velocity of gases in the annular space at this rate of working averaged about 330 feet per second.

The pressure drop of gases through the annular space was the equivalent of more than 7 inches of water. To move this large volume of gases against this pressure about 13.5 actual horsepower were required to drive the exhaust fan. Allowing 4 pounds of coal per horsepower hour, the coal consumption by the fan would be $13.5 \times 4 = 54$ pounds of coal per hour, or about 4.5 per cent of the total coal consumed. Although this amount of fan work may at first thought seem large, it compares favorably with the fan work re-

quired to operate boilers at their present low rates of working.

In 1905 Prof. Nicolson made some experiments with a small laboratory apparatus for the purpose of studying the laws of heat flow through metal from gas to water. These experiments were made at the works of Joseph Adamson & Co. The apparatus used, shown diagrammatically in figure 45, consisted mainly of two concentric tubes. Through the inner tube cold water flowed up; through the annular space, formed between the inner and the outer tubes, compressed (hot) air was forced down, so that the streams

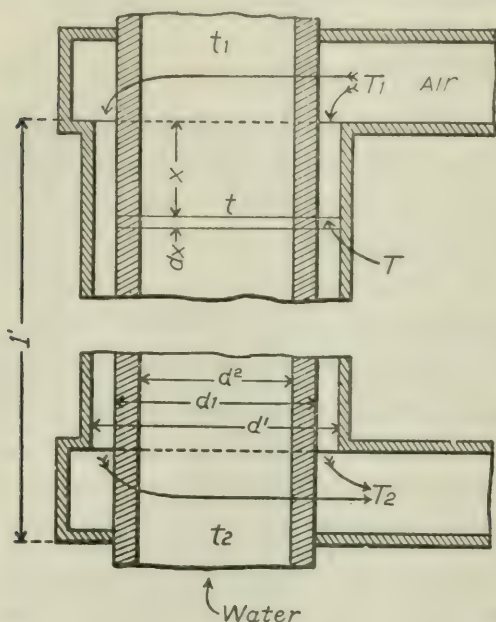


FIGURE 45.—Diagrammatic representation of Prof. Nicolson's laboratory apparatus.

of water and air flowed in opposite directions. The pressure of the air at the entrance into the annular space was about 60 pounds per square inch; it dropped 10 to 20 pounds while passing through the space. Its temperature at the entrance was between 100° and 160° F., and was 70° to 90° F. at the exit. The cold water entered at a temperature of 60° to 80° F.

At the entrance pressure and temperature the air was probably saturated with moisture. Any water that was precipitated during the abstraction of heat was collected in a vessel at the bottom of the apparatus and was measured. The relative humidity of the air leaving the collecting vessel was indicated by wet and dry bulb thermometers in an auxiliary glass box.

The quantity of air flowing through the annular space was regulated by varying the entrance pressure and also by varying the size of the exit orifice into the atmosphere.

Radiation from the outside of the air pipe was so small as to be almost negligible.

With this apparatus Prof. Nicolson has been able to obtain heat-transmission equivalent to the evaporation of over 10 pounds of water, from and at 212° F., per square foot of heating surface per hour, although the gas averaged only about 30° F. hotter than the metal of the inner tube. This rate of heat transmission is about three times the present average rate of heat transfer in steam boilers.

Prof. Nicolson states that the results of his heat-transfer experiments with the laboratory apparatus may be very concisely expressed by the following formulas:

(A) For heat transfer from air to pipe

$$Q_1 = H_1(T - \theta) = 3\rho_1 u_1(T - \theta), \quad (\text{I})$$

where Q_1 = B. t. u. transmitted per hour per square foot of area of pipe surface on the air side,

T = average air temperature (degrees Fahrenheit),

θ = average pipe temperature (degrees Fahrenheit),

ρ_1 = average density of air (pounds per cubic foot),

u_1 = average speed of air (feet per second),

$H_1 = 3\rho_1 u_1$ = coefficient of heat transfer per degree Fahrenheit difference.

(B) For heat transfer from pipe to water

$$Q_2 = H_2(\theta - t) = 6\rho_2 u_2(\theta - t), \quad (\text{II})$$

where Q_2 = B. t. u. transmitted per hour per square foot of area of pipe surface on the water side,

t = average water temperature (degrees Fahrenheit),

θ = average pipe temperature (degrees Fahrenheit),

ρ_2 = density of water (62.4 pounds per cubic foot),

u_2 = speed of water (feet per second),

$H_2 = 6\rho_2 u_2$ = coefficient of heat transfer per degree Fahrenheit difference.

When eliminating θ , the temperature of the metal, and estimating the constants for the most common values of $\frac{\rho_2 u_2}{\rho_1 u_1}$ that occur in practice, a rough approximation of the expression for heat transfer is

$$Q = 2.75\rho_1 u_1(T - t). \quad (\text{III})$$

Equation (III) states that, approximately, the heat transferred from a gas to water through a metal wall varies directly as the density of the gas, directly as the velocity of the gas over the dry surface of the tube, and directly as the temperature difference between the hot gas and the water. This is the same approximate law that Prof. John Perry derives in his treatment of this subject (see pp. 91-92 of this bulletin). The constant 2.75 in Prof. Nicolson's

equation is approximately right only for results obtained with his particular apparatus; for a different arrangement of the heating surfaces the constant would have to be modified.

The laboratory experiments of Prof. Nicolson, described in the preceding paragraphs, did not have a wide enough range of temperature. Therefore, Prof. Nicolson had another apparatus built at the Manchester Municipal School of Technology. The apparatus was substantially like the previous one, but air could be heated by passing it first through a coil over a gas burner. The pipes were about 12 feet long and of various diameters.

In 1906 and 1907 three distinct series of trials were made with this apparatus.

In the first series of experiments superheated steam was passed through the inner pipe and compressed air through the outer pipe. In the second series water was passed through the outer pipe (the annular space). In the third series preheated compressed air was used in the inner tube and cold water in the outer annular space. The weight of air was twice measured, first, by using the air compressor as a meter, and, second, by application of the laws of flow of air through sharp-edged circular orifices from a vessel in which the temperature and pressure of the air were observed. The steam in the first two sets of tests was condensed and weighed.

Prof. Nicolson says that the results are not yet available for publication, but that they afford satisfactory verification of Reynolds's great law.

PART IV.

GENERAL DISCUSSION.

MODES OF HEAT PROPAGATION.

It was stated in the introductory chapter that there are three modes of heat propagation. Each is entirely distinct from the other two, and the laws governing each differ essentially. In the process of heat absorption by boilers all of these three modes are active, either in combinations in which one predominates more or less, or acting singly in successive parts of the path of the heat travel. This is shown diagrammatically in figure 1. In order to study intelligently the problem of heat absorption by boilers, one must have a clear understanding of these three modes of heat propagation and of the laws which govern each. This understanding enables one to follow the path of heat travel and to apply the proper laws in the solution of the various parts of the problem. Perhaps the most direct cause of the present neglected state of the art of steam-boiler design has been the designer's lack of clear understanding of the modes of heat propagation. One must admit that the scientific aspects of the art of steam making are much neglected when he compares steam engineering with electrical engineering and remembers that the latter is much the younger. The electrical engineer has supplemented his knowledge liberally from the allied sciences of physics and chemistry. On the other hand, the greater number of boiler engineers, even in recent years, have had a fair understanding only of the law of heat conduction, and have often applied this law without discrimination in the solving of any heat-transfer problem. The reason for the general application of this law lies in its simplicity. The law of radiation was more complex and usually unknown. The law of convection was only partly known, even to physicists.

In the following paragraphs each of these laws is discussed so far as it is known with regard to its special application to the steam-boiler proposition.

CONDUCTION.

In the discussion of figure 1 it was stated that from the dry surface of the heating plate of a boiler the heat is transmitted through the metal and its coatings to the wet surface of the plate by conduction only. The quantity of heat that can be transmitted through a unit area of the plate in a unit of time depends on the difference of the

temperatures and the conductivities of the substances between the dry and the wet surfaces of the heating plate. This law can be expressed by the following simple formula:

$$(18) \quad H = \frac{C}{d}(t_1 - t)$$

in which H = the quantity of heat transmitted per unit of area of the heating plate,

C = the average conductivity of the substances between the dry and wet surfaces,

d = the distance between the two surfaces,

t_1 = the temperature of the dry surface, and

t = the temperature of the wet surface.

The law governing the rate of heat transmission by conduction is the same as Ohm's law, much used in electric-conduction problems.

It is evident that with any physical condition of the plate and the surfaces the rate of heat transmission depends entirely on the excess of the temperature of the dry surface over that of the wet surface. For instance, if it is required to transmit double the quantity of heat in the same length of time, the difference between the temperatures of the two surfaces must be doubled. Since the temperature of the wet surface is nearly the same as that of the steam in the boiler, and therefore can not be lowered, the temperature of the dry surface must be raised; since it is this dry surface of the heating plate that cools the furnace gases, raising its temperature raises the temperature of the escaping gases. Thus one can see that, with the same conditions of the heating plate and the same initial temperature of the furnace gases, the temperature of the escaping gases will rise somewhat with increasing capacity.

To illustrate the law of heat conduction, a specific case is here assumed and the factor $(t_1 - t)$ for ordinary boiler capacity computed by the use of formula (18). With the same formula also is computed the probable rise in flue-gas temperature when the capacity of the boiler is raised.

The heat conductivity of iron at 400° F., given in the Smithsonian Physical Tables, is about 0.0005. This means that if the two surfaces of a steel plate 1 inch thick are kept at a temperature difference of 1° F., every square inch of the plate will transmit 0.0005 B. t. u. per second; if the temperature difference is 10° F., 0.005 B. t. u. will be transmitted; or, if the thickness of the plate is 0.1 inch and the temperature difference of the surfaces is 10° F., each square inch will transmit 0.05 B. t. u. per second.

The walls of the tubes of a water-tube boiler are about 0.1 inch thick; the tubes of a locomotive are often thinner.

At the rate of 10 square feet of heating surface per boiler horsepower the heat transmitted per square inch per hour is

$$\frac{34.5 \times 965}{10 \times 144} = 23.1 \text{ B. t. u.}$$

or

$$\frac{23.1}{60 \times 60} = 0.0064 \text{ B. t. u. per second.}$$

To transmit this quantity of heat through the walls of the tube 0.1 inch thick requires a temperature difference between the two surfaces of

$$\frac{0.0064}{0.005} = 1.3^\circ \text{ F.}$$

Now, this is a surprisingly small temperature difference and shows that the resistance of the metal to heat transfer is very low. Undoubtedly something else than the conductivity of the metal is to blame for the low rate of steam production in steam boilers. Perhaps it may be presumed that the soot and scale coatings generally present (though they may be very thin) add to the resistance of the plate to heat transfer. Suppose that in a well-kept boiler the resistance to heat passage of the soot and scale is ten times that of the metal alone; for the ordinary rate of working there is then a temperature drop through the soot and scale of 13° F. ; this, with the drop of 1.3° F. through the metal, makes a total temperature drop of 14.3° F. between the dry and the wet surface of the tube. Even this is a small temperature drop, and indicates possibilities of working the tubes at higher rates. Thus, to make 1 boiler horsepower on 1 square foot of the tube heating surface, which is ten times the ordinary rate of working a boiler, the temperature difference between the wet and the dry surface of the tube need be only 143° F. This figure indicates that even for the high rate of working assumed, the temperature drop between the two surfaces of the tube is only a small fraction of the total drop between the moving gases and the boiler water. Figure 46 shows relatively the average temperature drop between the hot moving gases and the boiler water when the initial temperature of the gases is $2,500^\circ \text{ F.}$ and the boiler is working at ten times the usual rate of steam making. In the figure the temperature of the boiler water is assumed to be 350° F. The temperature drop from the wet surface to the boiler water is estimated to be 10° F. , making the temperature of the wet surface 360° F. The drop through the metal itself has been figured to be 13° F. The drop from the dry surface to the metal and that from the metal to the wet surface are both assumed to be one-half of 130° F. , although it is very probable that the drop on the dry side of the metal is much higher than that on the wet side. The total drop from the dry surface to the boiler water is then 153° F. , making the tem-

perature of the dry surface 503°F . If the temperature of the escaping gases is 600°F ., then the average temperature of the moving gases is approximately $\frac{2,500 + 600}{2} = 1,550^{\circ}\text{F}$. Consequently there are the following temperature drops in the path of heat travel: From the hot moving gases to the dry surface, $1,550 - 503 = 1,047^{\circ}\text{F}$.; from the dry to the wet surface, $503 - 360 = 143^{\circ}\text{F}$.; and from the wet surface to the water in the boiler, $360 - 350 = 10^{\circ}\text{F}$.

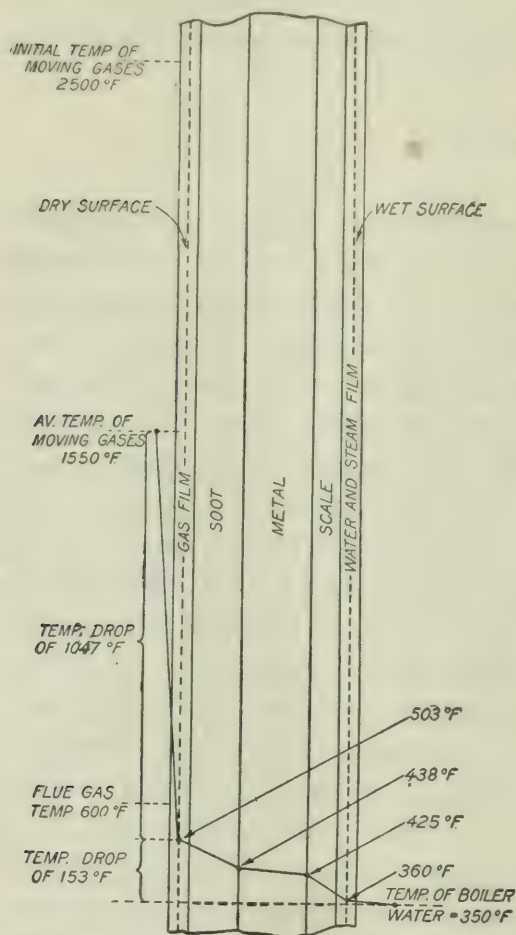


FIGURE 46.—Diagram of temperature drop from moving gases through the heating plate to boiler water.

States Geological Survey indicate that in the lower row of tubes in a Heine boiler the maximum temperature difference between the wetted surface of a tube and the water in the center of the tube is probably less than 15°F . when the boiler is working far above its rated capacity, and the combustion-chamber temperature is about $3,000^{\circ}\text{F}$. If these measurements are correct, they indicate that natural circulation of the water will be sufficient in boilers working at several times the present-day rate, provided the boilers are suitably designed.

The foregoing consideration show that the process of getting the heat to the dry surface is slow and that it will have to be hastened if the rate of working steam boilers is to be increased. As shown in figure 1, the heat enters the dry surface by radiation and convection,

It is true that the values given above are either estimated or figured, nevertheless observation indicates that they are very close to the facts. One can say with a high degree of certainty that the cause of the low rate of heat transmission in the present boilers lies in the transmission from the hot gases to the dry surface of the tube. The metal itself, even with a moderate coating, is capable of conducting enough heat to the water with a comparatively small part of the total temperature drop between the gases and boiler water. On the inside of the boiler tube the water, on account of its high heat capacity, readily takes the heat from the surface of the tube so that the temperature drop usually needed is very small. It may be stated here that some determinations made by the technologic branch of the United

and it is therefore these two modes of heat propagation that must be studied in order that every factor in the laws governing them may be utilized to the best advantage.

THE RATE AT WHICH HEAT IS IMPARTED BY RADIATION.

The quantity of heat which a boiler receives by radiation in a unit of time may be taken to be approximately proportional to the difference between the fourth powers of the absolute temperatures of the hot parts of the furnace and the sooted surface of the boiler plate. This law of radiation is known as Stefan and Boltzmann's Law.^a

About 30 years ago Stefan deduced this law from the results of experiments, and some years later Boltzmann demonstrated mathematically that from thermodynamic considerations the fourth-power law should hold exactly for an ideal black body. There are, however, no substances in nature that absorb and radiate heat exactly as the black body does. The sooted surface comes close to it (within 4 or 5 per cent, depending on the kind of soot), and can be taken for all practical purposes as a standard. Since, in the steam-boiler problem, the heat exchange by radiation is between partly sooted surfaces, the law can be applied without any serious error. The radiation law is expressed by the following equation:

$$(19) \quad H = C(T^4 - t^4).$$

Where H = the net heat exchanged between the hot and the cold surface per unit of the hot surface per unit of time;

T = the absolute temperature of the hotter surface;

t = the absolute temperature of the colder surface, which surrounds the hot surface;

and C = a constant depending on the unit of area and time, on the unit in which the heat is measured, and on the scale in which the temperatures are expressed.

If H is expressed in small calories per square centimeter of the hot surface per minute, and if T and t are expressed in degrees centigrade on the absolute scale, then

$$(20) \quad C = 7.65 \times 10^{-11} = \frac{7.65}{100,000,000,000}$$

If H is expressed in B.t.u. per square foot of the hot surface per minute, and T and t are expressed in degrees Fahrenheit on the absolute scale, then

$$(21) \quad C = 2.66 \times 10^{-11} = \frac{2.66}{100,000,000,000}$$

^a For a detailed discussion of this law, see Warder, C. W., and Burgess, G. K., Optical pyrometry: Bull. 2, Bureau of Standards, 1905.

These constants are good only in case the hotter surface is completely surrounded by the cooler surface, or, in case the conditions are such that the hot surface can not "see" anything but the cold surface. Cases A and B of figure 47 illustrate two typical instances in which this condition is fulfilled.

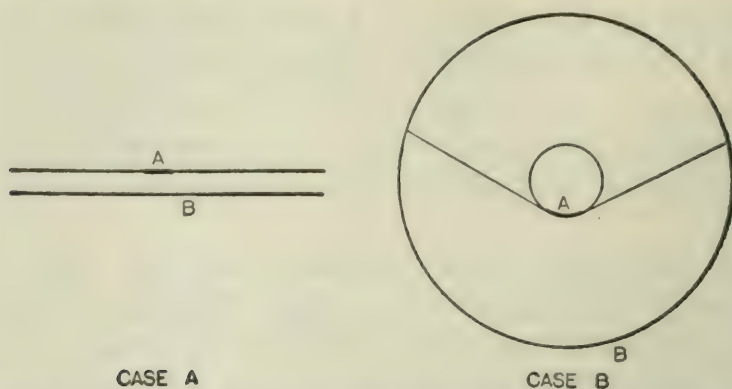


FIGURE 47.—Cases in which the fundamental conditions of Stefan and Boltzmann's radiation law are entirely fulfilled.

In figure 47, Case A, the hot surface A is a plane; the cold surface B is also a plane, parallel to the first one and infinitely large, so that any portion of the hot surface A can not "see" anything but the cold surface B. That is, the hot surface A is completely exposed to the cold surface B, or the angle of exposure of the hot surface approaches the spherical angle of 180° .

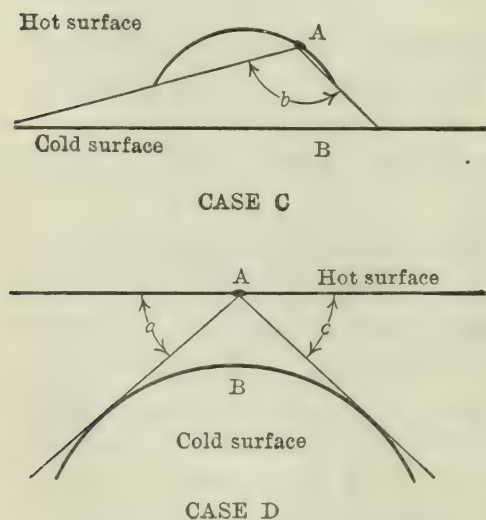


FIGURE 48.—Cases in which the fundamental conditions of Stefan and Boltzmann's radiation law are imperfectly fulfilled.

In figure 47, Case B, the hot surface A is the outside surface of a spheroidal body that is inclosed in a larger hollow spheroid, the inside surface of which forms the cold surface B. In this case any portion of the hot surface can "see" nothing but the cold surface; that is, the hot surface is completely exposed to the cold surface B.

In figuring the net heat exchanged between the hot and the cold surface in either of the two cases illustrated, the constant C can be applied directly in the above formulas.

In figure 48 are illustrated two typical cases where the hot surface is not completely exposed to the cold surface and where the constant C can not be applied directly. In Case C the hot surface A is exposed not only to the cold surface B, but also to other parts of the hot surface itself. In Case D the hot surface A is exposed partly to the cold surface B and partly to the open space outside the cold surface. That is, the heat passing from A through angles a and c misses the

cold surface entirely. In the last two cases the constant C must be multiplied by the solid angle of exposure expressed as a percentage of the spherical angle of 180° .

The locomotive furnace presents the simplest practical problem in radiation. In it the hot surface is replaced by the fuel bed and the cold surface by the tube sheet, the crown sheet, the front sheet, and the plates on both sides of the fuel bed. As shown in figure 49, the only other surface exposed to the fuel bed is the fire door, which subtends

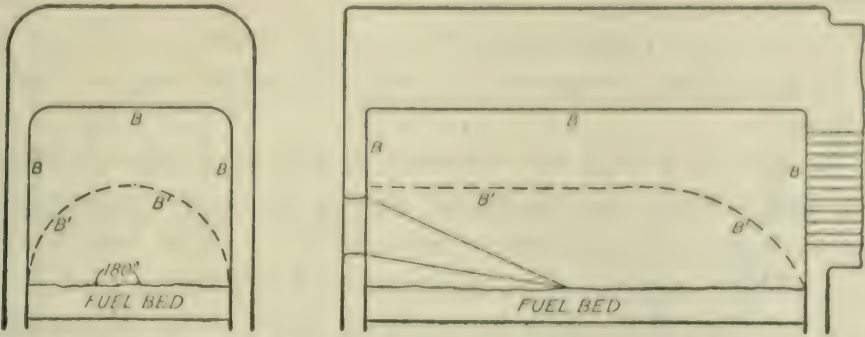


FIGURE 49.—Application of Stefan and Boltzmann's radiation law to a locomotive boiler.

a very small angle of the fuel bed's "vision" and can be neglected. It should be stated in connection with the locomotive furnace that when the temperatures remain constant the amount of heat received by the boiler by radiation depends on the extent or area of the fuel bed (when the top of the fuel bed is approximately a plane surface) and not on the exposed area of the boiler's heating surface. That is, referring to figure 49, the amount of heat radiated to the boiler would be the same if the inside of the fire box had the shape shown by the dotted lines B' .

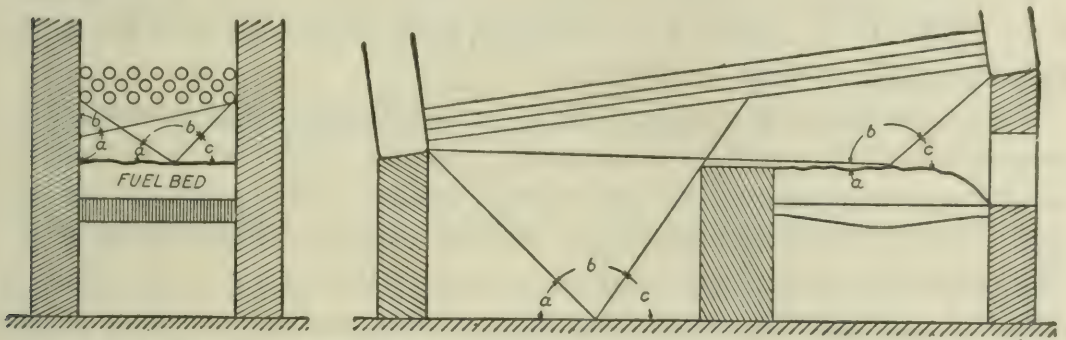


FIGURE 50.—Application of Stefan and Boltzmann's radiation law to an externally fired boiler.

In furnaces of externally fired boilers the problem of heat radiation is a more complicated one. On the assumption that the temperature of the inside of the walls of the furnace is the same as that of the fuel bed, the fuel bed and the inside of the furnace walls may be designated the hot surface. Under these conditions any portion of the hot surface will be exposed not only to the cold surface of the boiler, but also to other parts of the hot surface itself. This is illustrated in figure 50; any portion of the hot surface is exposed to the boiler through the

solid angle b and to other parts of the hot surface itself through the angles a and c .^a To obtain the net amount of heat radiated by each portion of the hot surface to the boiler, constant C in equation (19) would have to be multiplied by the solid angle b of exposure expressed as a fraction of a spherical angle of 180° . In any special case of this type it would not be difficult to get the approximate average angle of exposure of the fuel bed, side walls, and the rear and front door by averaging the angles of exposure of four or five small portions of each of the hot surfaces.

It is apparent from equation (19) that the quantity of heat a boiler receives by radiation increases very rapidly as the temperature of the furnace rises.

Let it be assumed that two boilers, A and B, both of similar construction and setting, are working under the same pressure, corresponding to a steam temperature of 350° F. What are the relative quantities of heat imparted to each boiler if the furnace of A is operated at $3,040^\circ$ F. and the furnace of B at $2,040^\circ$ F.? These temperatures agree with observations actually made. Assuming the temperature of the soot coating to be about 540° F., the relative quantity of heat received by A is $2.54 \times 10^{-11}(3,500^4 - 1,000^4) = 3,785$ B. t. u. and the heat received by B is $2.54 \times 10^{-11}(2,500^4 - 1,000^4) = 966$ B. t. u. Therefore, boiler A receives $\frac{3,785}{966} = 3.92$ times as much heat as boiler B.

In the above example the temperature shown by the ordinary Fahrenheit scale is changed to absolute temperature by adding to the former 460° .

The two examples indicate that by raising the furnace temperature from $2,040^\circ$ F. to $3,040^\circ$ F. the rate of heat impartation to the boiler by radiation is nearly quadrupled.

Figure 51 shows graphically the relation between the furnace temperature and the quantity of heat imparted to a boiler by radiation, for constant temperatures of the soot coating. The abscissæ of the chart are furnace temperatures of the absolute Fahrenheit scale. The ordinates, when used with the scale on the left, give the B. t. u. radiated per minute per square foot of hot surface exposed to the boiler surface only; when used with the scale on the right the ordinates give equivalent evaporation per hour for the same area of the hot surface. Each curve gives these values for one constant temperature of the sooted surface.

Thus, for example, let it be assumed that the furnace temperature is $3,000^\circ$ F., absolute ($2,540^\circ$ on the ordinary Fahrenheit scale), and

^a For simplicity it is assumed that no tiles are around or between the lowest row of tubes anywhere along their length; in other words, that all of the roof above the fuel bed and combustion chamber is cold surface, at boiler-water temperature.

the temperature of the sooted surface of the plate is 800° F., absolute (340° on the ordinary Fahrenheit scale); then, by following the vertical line of $3,000^{\circ}$ F. furnace temperature to its intersection with the curve denoted as constant temperature of sooted surface of 800° F., absolute, one obtains the length of the ordinate which, when used with the scale on the left, tells that 2,040 B. t. u. are radiated to the boiler per minute by each square foot of totally exposed hot surface. The scale on the right indicates that at this rate of heat radiation each square foot of the totally exposed hot surface causes the evaporation of 126 pounds of water per hour.

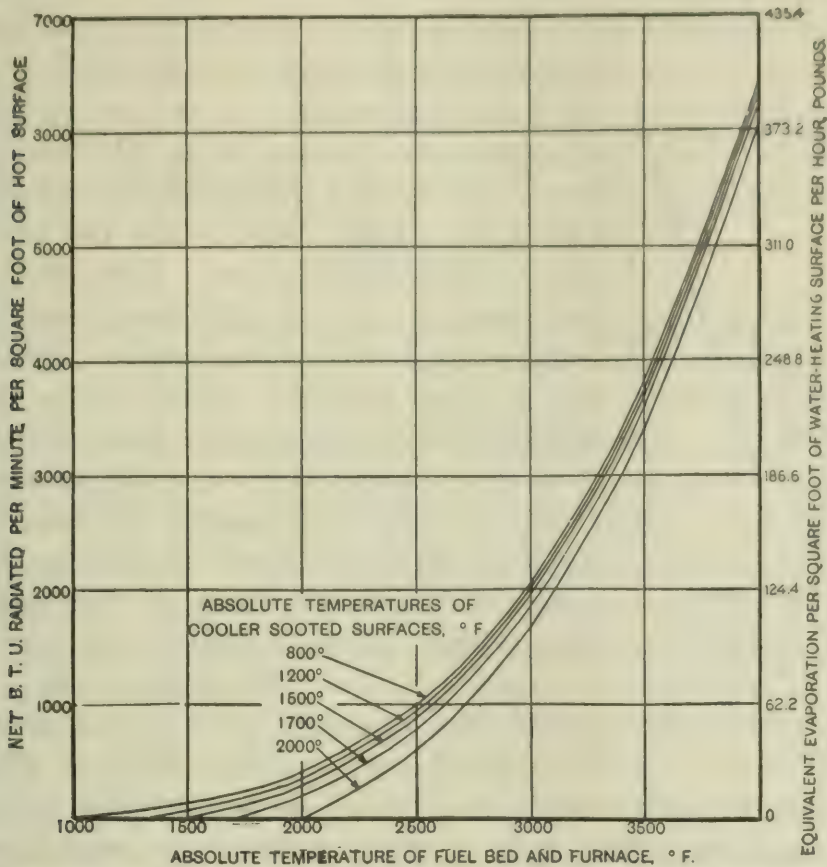


FIGURE 51.—Relation between furnace temperature and quantity of heat imparted to a boiler by radiation for several constant temperatures of the soot coating on the water-heating plates.

Again, let it be assumed that the furnace temperature is $3,100^{\circ}$ F., absolute ($2,640^{\circ}$ on the ordinary Fahrenheit scale), and the temperature of the sooted surface is $1,700^{\circ}$ F., absolute ($1,240^{\circ}$ on the ordinary Fahrenheit scale); then, by following the vertical line of furnace temperature of $3,100^{\circ}$ F. to its intersection with the curve denoted as constant temperature of sooted surface of $1,700^{\circ}$ F., absolute, one gets the length of the ordinate which, if referred to the scale on the left, shows that 2,120 B. t. u. are radiated per square foot of completely exposed hot surface, and if used with the scale on the right, shows that 130 pounds of water is evaporated per hour by every square foot of completely exposed hot surface.

These two examples show that although in the second case the temperature of the heating plate is 900° F. higher, and the temperature of the furnace is only 100° F. higher than in the first case, nevertheless more heat is received by the heating plate in the second case. It may be said, then, that a rise or drop in the furnace temperature is a great many times more influential than an equal rise or drop in the temperature of the sooted surface of the heating plate, so far as the absorption of heat by radiation is concerned.

In figure 51 the given rate of net heat radiation is per square foot of the totally exposed hot surface and not per square foot of the boiler-heating surface. It has been stated in connection with figure 48 that at the same temperatures the net quantity of heat exchanged

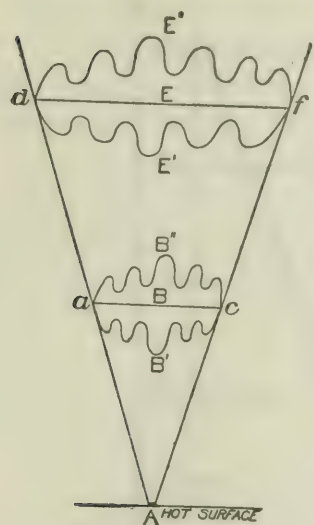


FIGURE 52.—Diagram illustrating the fact that shape and distance of cold surface do not affect the radiation of heat.

between the hot and the boiler surfaces depends on the extent of the hot surface and the angle of exposure of the hot surface to the boiler surface. Figure 52 further illustrates this feature. Let A be a small unit of the hot surface and aBc be the cold surface. Now, the cold surface does not receive any more heat from the small area A of the hot surface, even though the former has the shape $aB'c$, or $aB''c$, although in the latter two cases the cold surface may have twice as much area as in the first case. Again, if the cold surface dEf , having four times the area of aBc , be placed at twice the distance from A , the quantity of heat received from A will be the same as was received by the cold surface aBc . Neither would the quantity of heat be increased if corrugated cold surfaces $dE'f$ or $dE''f$ were

substituted for dEf . From this illustration it will be noticed that radiant heat resembles light in every respect—a resemblance that is confirmed by physicists.

In the preceding discussion of radiation in boiler furnaces, only the fuel bed and furnace walls have been considered as radiating surfaces. In reality the luminous flames perhaps radiate considerable heat and therefore increase the radiating surfaces. However, as the flames vary in size and shape with different fuels, rate of combustion, and supply of air, it is difficult to figure their effect exactly. In any special case the average additional area and angle of exposure of the flames can be estimated and added to the area of the hot parts of the furnace. The object of this discussion is to elucidate the principles of radiation and to suggest methods of general analysis of special cases of radiation in boiler furnaces.

The substance of the foregoing discussion of the law of heat radiation may be summarized in the following brief statements:

The quantity of heat imparted to a boiler by radiation from the hot surfaces of the furnace and fuel bed depends (*a*) on the extent of the hot surfaces of the furnace and the angle of exposure to the cold surfaces of the boiler; (*b*) on the difference between the fourth powers of the absolute temperatures of the "hot" and "cold" surfaces.

In any given steam-generating apparatus the first of the above conditions is fixed, and therefore can not be used to increase the output of the boiler. As to the second, it may be said that in well-operated plants the furnace temperatures can not be raised much higher; so that there is little hope of raising the capacity of boilers by increasing the amount of heat imparted to them by radiation. Furthermore, in a modern water-tube boiler most of the total heat absorbed is imparted to the boiler's surfaces by convection, so that increasing the rate of radiation would raise the capacity of the boiler but little. Much more is to be expected from the possibilities of increasing the rate of heat impartation to boilers by convection.

CONVECTION.

In modern steam boilers by far the greater percentage of the total heat absorbed by the heating surface is imparted to it by convection. Furthermore, the gases, as they pass along the heating surfaces, are cooled by convection, and therefore the degree of cooling of the gases before they finally leave the boiler is dependent solely on the activity of convection. Consequently, convection largely determines the efficiency of boilers. It was particularly with the object of studying the laws of heat transmission by convection that the authors undertook the laboratory experiments (with small boilers) that form the nucleus of this bulletin.

In the section on "How heat flows into the boiler water" (p. 18) it has been said that as the fuel burns in the boiler furnace the heat evolved is largely absorbed by the gaseous products of combustion. These gases then flow over the heating surface of the boiler, to which they impart some of the heat they contain. This impartation of heat takes place almost entirely by convection; that is, the molecules of the flowing gases come in contact with the dry surface of the boiler plate and impart their heat to it. The rate at which the heat is imparted to a unit of surface depends on the number of gas molecules coming in contact with it.

This law is very clearly expressed in Prof. Reynolds's paper by the following statements:

The heat carried off by air or any other fluid from a surface,^a apart from the effect of radiation, is proportional to the internal diffusion of the fluid, at and near the surface; that is, is proportional to the rate at which particles or molecules pass backward and forward from the surface to any given depth within the fluid. * * *

^a In the case of a steam boiler the heat is carried by the gas to the surface.

Now, the rate of this diffusion has been shown from various considerations to depend on two things:

1. The natural internal diffusion of the fluid when at rest.
2. The eddies caused by visible motion, which mixes the fluid up and continually brings fresh particles into contact with the surface.

These few sentences from Prof. Reynolds's paper form the foundation of the law governing the rate of heat transmission by convection and deserve careful study. In his paper Prof. Reynolds expressed this law of heat transmission by convection by the equation

$$(1) \quad H = A(T - t_1) + Bqv(T - t_1),$$

where H = the quantity of heat imparted by the gas to a unit surface of the metal,

$(T - t_1)$ = the temperature difference between the metal surface and the gas,

q = the density of the gas,

v = the velocity of the gas over the metal surface; and

A and B are constants depending on the nature of the gas.

The first term on the right side of the equation expresses the effect of the natural diffusion of the gas; the value of this term depends only on the difference of temperature between the gas and the metal; the second term expresses the effect of diffusion caused by the motion of the gas; the value of this term depends on the density of the gas, the velocity of the gas, and on the difference of temperature between the gas and the metal surfaces. The validity of the first term will perhaps be generally admitted because it contains the temperature factor only. The second term, however, on account of the density and the velocity factor may appear somewhat obscure. The following explanation may help one to understand it:

The rate of heat impartation by a stream of flowing gas to a metal surface, as has been stated, is directly proportional to the rate at which molecules of gas pass forward and backward from the stream to the metal surface. Now, the density of a gas is proportional to the number of gas particles, or molecules, in a unit of volume. It is reasonable, then, to expect that when there are more particles of gas in a unit of volume next to a metal surface proportionately more gas particles, or molecules, will hit or come in contact with a unit of that surface. The density of a gas at constant pressure varies inversely as the absolute temperature; that is, the density decreases as the absolute temperature rises. On this account there is a partial neutralization of gain when striving for high temperatures in steam boiler practice, for, as the temperature is raised, the number of molecules in action against any unit of the dry surface of the heating plate is reduced.

In considering the velocity factor let the reader form a mental image of the appearance of an intensely magnified cross section of

the dry surface of the heating plate. Entangled among the particles of the soot are particles, or molecules, of gas held close together by the attraction of the solid and forming a dense film next to the layer of soot—or the metal in case the boiler is clean; farther out the molecules are wider apart. All these molecules of gas, from those entangled among the particles of the soot to those in the gas of normal density, are to be regarded as in an invisibly rapid state of vibration; however, those close to the soot are more or less bound by the attraction of the solid soot or metal.

Now, a gas is a very poor conductor of heat, and if dependence were placed solely on its conduction the process of transferring heat from the moving gas to the adhering film would be very slow indeed. The only quick way of getting heat into the adhering film is to dislodge the cold (or slowly vibrating) molecules from this adhering film of gas and replace them by hot (rapidly vibrating) ones from the body of hot gas. This replacement is effected partly by the natural diffusion of the gas—first term of equation (1)—but mostly by the velocity of the gas. It can be imagined that as a stream of hot gas moves along the adhering film, an average of one-third of its molecules vibrate back and forth perpendicularly to the surface and dislodge or displace the molecules in the adhering film. The faster the gas moves the greater the number of molecules or particles of gas that pass each square inch of the heating surface in a unit of time, and consequently the greater the number of hot molecules that will hit or penetrate into the adhering gas film and replace cold molecules of gas. Thus the dislodging effect of these moving molecules on the molecules of the adhering film is proportional to the velocity of the mass of the gas moving over the heating plate. It is this dislodgment that makes a boiler respond in quantity of steam made to any reasonable demands put on it.

By referring to Prof. Reynolds's equation (1), one can see that if the velocity of the gas continues to increase the second term on the right hand of the equation becomes large and the effect of the first term so small in comparison with that of the second term that beyond low velocities of the gas the first term may be ignored and the equation reduced to equation (7) derived by Prof. Perry:

$$(7) \quad H = Cqv(T - t_1).$$

As t_1 , the temperature of the dry surface, is generally not much higher than t , the temperature of the wet surface or the boiler water, we shall at first assume for the sake of simplicity that

$$T - t_1 = T - t.$$

Designating $T - t$ by θ , equation (7) can be written

$$(22) \quad H = Cqv\theta.$$

This is a simpler equation and when used with discrimination it is fairly accurate for most practical cases.

RELATION BETWEEN THE LENGTH OF GAS PATH AND THE EXIT TEMPERATURE OF GASES.

The simplicity of equation (22) is more apparent when it is used as a basis for finding the relation between the initial temperature of the gases and their temperature at any point along their path over the heating surface of a boiler. This relation is found in the following way:

Let w be the weight of gases passing over the heating surface of a boiler per second, and x the distance of any small area under discussion from the furnace end of the boiler. Then the heat lost by the gases during passage along any small part of the gas path (the "small area" just mentioned) is equal to the heat gained or absorbed by the heating surface of that part of the gas path, or

$$(23) \quad -swd\theta = C'qv\theta dx.$$

In this equation s is the specific heat of gases at constant pressure, $d\theta$ is the temperature drop of the gases while passing over a small part, dx , of their path. Since the heat is abstracted from the gases, the sign on the left side of the equation is minus ($-$).

Now, the density of gases varies inversely as their absolute temperature; therefore

$$q = \frac{c}{\theta + K},$$

where K is the absolute temperature of the boiler water and c some constant.

The velocity of the gases varies directly as their volume, and their volume varies directly and jointly as their absolute temperature and the weight to be passed over the heating surface; therefore

$$v = c'w(\theta + K),$$

where K is as before the absolute temperature of the boiler water and c' some constant.

Substituting these values for q and v in equation (23), we get

$$(24) \quad -s \frac{wd\theta}{\theta} = Cwdx,$$

where C is a new constant equal to $C' \times c \times c'$.

Integrating (24) we get

$$(25) \quad -sw \log \theta = Cwx + k.$$

The constant of integration, k , can be determined from the relation that when $x=0$, $\theta=\theta_0$, the initial temperature; and therefore

$$k = -sw \log \theta_0.$$

Substituting for k in (25) we have

$$\begin{aligned} -sw \log \theta + sw \log \theta_0 &= Cwx, \text{ or} \\ -sw \log \frac{\theta}{\theta_0} &= Cwx \\ -\log \frac{\theta}{\theta_0} &= \frac{Cwx}{sw} = cx, \text{ where } c = \frac{C}{s}. \end{aligned}$$

Therefore $\frac{\theta}{\theta_0} = e^{-cx}$, or

$$(26) \quad \theta = \theta_0 e^{-cx}.$$

In equation (26), if x is equal to the full length of the gas path, θ will be the temperature of the escaping gases above the boiler-water temperature.

With Prof. Reynolds's equation (1) containing the two terms on the right side as bases, the relation between θ and θ_0 can be found in a similar way. In this case it is assumed that $(T - t_1) = (T - t) = \theta$. Thus with the same reasoning one obtains

$$(27) \quad -swd\theta = (\Lambda\theta + Bcw\theta)dx.$$

$$(28) \quad -\frac{swd\theta}{\theta} = (\Lambda + Bcw)dx.$$

Integrating and determining the constant of integration we have

$$(29) \quad -\log \frac{\theta}{\theta_0} = \frac{\Lambda x + Bcw x}{sw}, \text{ or}$$

$$\theta = \theta_0 e^{-\frac{\Lambda x + Bcw x}{sw}}$$

According to equation (26), for any given boiler θ depends only on θ_0 , the initial temperature; but according to equation (29), θ depends also on the weight of gases passing over the heating surface of the boiler.

In the derivation of equations (26) and (29) it was assumed that the temperature of the dry surface was the same as that of the steam in the boiler, but we know from the law of heat conduction that there must be a temperature difference between the two surfaces in order that heat shall flow through the metal plate and its coating. The relation between the temperature of the dry and wet surfaces is expressed by equation, the character c being used for C (18):

$$\begin{aligned} H &= \frac{c}{d}(t_1 - t), \text{ or} \\ t_1 &= \frac{Hd}{c} + t. \end{aligned}$$

Substituting this value for t_1 in equation (7) we get the correct expression for H :

$$H = Cqv \left(T - \frac{Hd}{c} - t \right).$$

Substituting for q and v their value in w , simplifying, and making $\theta = (T - t)$ as before, we have

$$(30) \quad \begin{aligned} H &= \frac{C'w\theta}{c + Cwd}, \text{ or,} \\ -swd\theta &= \frac{C'w\theta}{c + Cwd} dx. \end{aligned}$$

Integrating and determining the constant of integration we obtain the relation

$$(31) \quad \theta = \theta_0 e^{-\frac{Cx}{c + Kw}}$$

in which C , c , and K are constants. Equation (30) states that the temperature of gases at any point along their path depends on the initial temperature and also on the weight of gases passing along the path.

Using as a basis the equation $H = A\theta + Bqv(T - t_1)$, and also the relation $t_1 = \frac{Hd}{c} + t$, between the temperature of the dry and wet surfaces, the relation between the initial temperature of the gases and their temperature at any point of their path is similarly found to be

$$(32) \quad \theta = \theta_0 e^{-\frac{(Ac + Bkw)x}{(c + Ad + Bkw)w}}$$

where A and B are the same constants as in equation (1), c the average conductivity of the heating plate, including all materials between the hot and the wet surface, d the distance between the surfaces, and k some constant to be determined that depends on the specific heat and some other physical properties of gases.

Equation (32) states that for any given boiler the temperature of gases at any point x of the gas path depends on the initial temperature of the gases and on the weight of gases passing over the heating surface in a unit of time. If x is equal to the length of the gas path, then equations (26), (29), (31), and (32) give the temperature of the escaping gases.

RELATION BETWEEN THE TRUE BOILER EFFICIENCY AND THE WEIGHT OF GASES PASSING THROUGH A BOILER.

The temperature of the escaping gases being determined, the true boiler efficiency can be determined, as follows:

Let θ_1 = the temperature of the escaping gases above the temperature of the boiler water, and

θ_0 = the initial temperature of the gases above boiler-water temperature.

Then the heat available for the boiler is proportional to θ_0 , and the heat absorbed by the boiler proportional to $(\theta_0 - \theta_1)$. The true boiler efficiency E_4 is expressed by the general equation

$$(33) \quad E_4 = \frac{\theta_0 - \theta_1}{\theta_0}.$$

By substituting for θ_1 its values as determined by means of equations (26), (29), (31), and (32) the following respective expressions for the true boiler efficiency are obtained:

$$(34) \quad \text{From (26)} \quad E_4 = 1 - e^{-cl}$$

$$(35) \quad \text{From (29)} \quad E_4 = 1 - e^{-\frac{Al + Bcw}{sw}}$$

$$(36) \quad \text{From (31)} \quad E_4 = 1 - e^{-\frac{Cl}{c + Kw}}$$

$$(37) \quad \text{From (32)} \quad E_4 = 1 - e^{-\frac{(Ac + Bkw)L}{(c + Ad + Bkw)w}}$$

When the last four equations with their proper constants are plotted with E_4 as ordinates and w , the weight of gases, as abscissæ, the resulting curves have shapes as shown in figure 53.

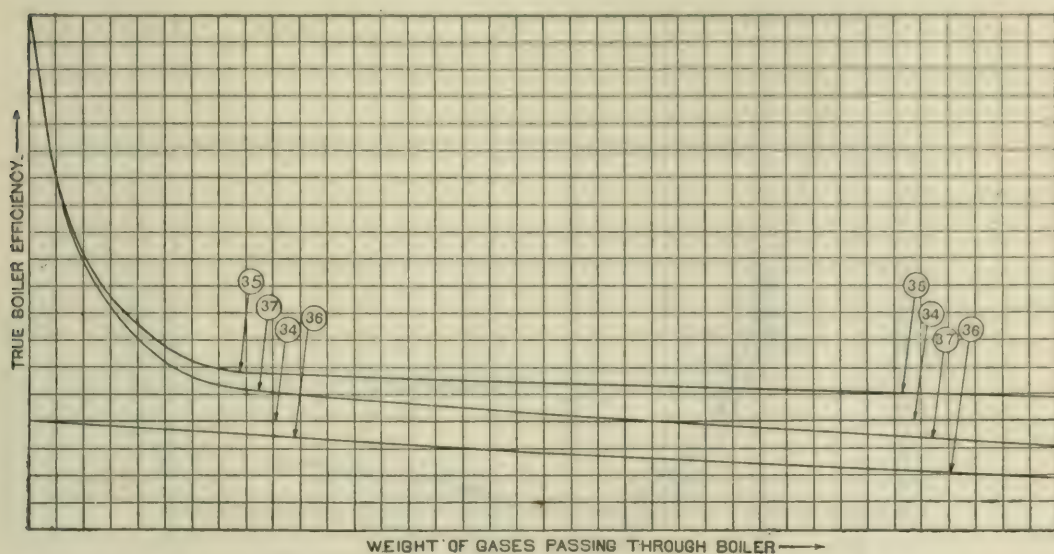


FIGURE 53.—General shapes of efficiency curves of equations (34), (35), (36), and (37).

In equation (34) E_4 is independent of the weight of gases, therefore the efficiency curve is a straight line parallel to the w axis as shown by curve (34).

The curve representing equation (35) with w as abscissæ is a logarithmic curve starting from a point of $E_4=100$ and $w=0$, and is asymptotic to the curve of equation (34). This latter property can be readily seen by examining the exponent of equation (35); if w is very large the first term Al will be so small in comparison with the second term that Al can be dropped out and the exponent will reduce to cl , the exponent of equation (34).

Equation (36) when plotted gives a logarithmic curve. It starts from the same point as the curve of equation (34) and is asymptotic to zero efficiency line. When w is zero, the term Kw in the exponent,

$-\frac{Cl}{c + Kw}$, reduces to zero and the exponent becomes the same as that of equation (17). When w is infinitely large the exponent reduces to zero and $e^{-\frac{Cl}{c + Kw}}$ becomes equal to 1 and therefore E_4 is zero.

The graph of equation (37) is a logarithmic curve starting from a point of $E_4 = 100$ when $w = 0$, and is asymptotic to zero efficiency line. When w is equal to zero the exponent becomes infinitely large and reduces $e^{-\frac{(Ac+Bkw)l}{(c+Ad+Bkw)w}}$ to zero; this makes the efficiency equal to 100 per cent. When w is very large the value of the exponent approaches zero.

The efficiencies obtained by equations (36) and (37) approach zero when the weight of gases becomes very large, because the equations take into account the conductivity of the plate, which, although great, is finite. When a large weight of gases is passed over the heating surface the quantity of heat imparted to the dry surface

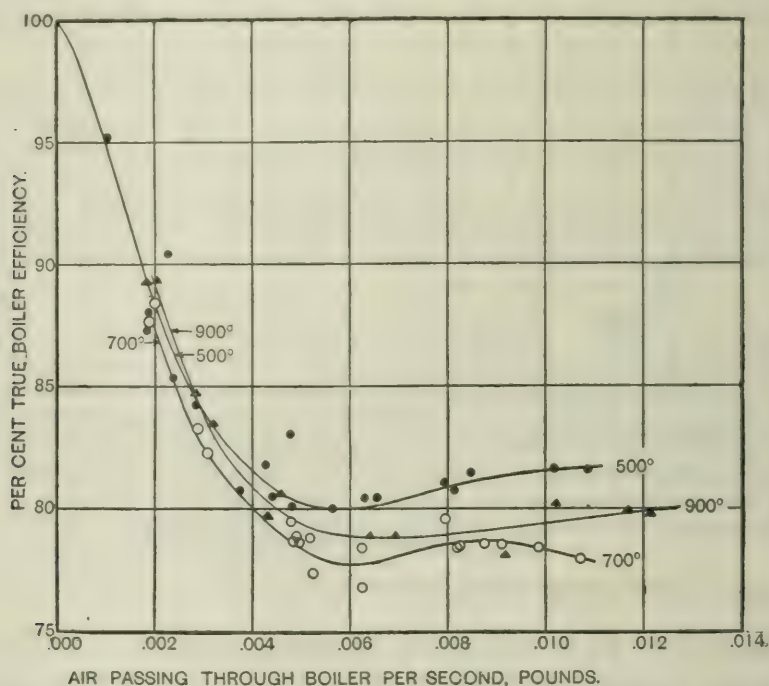


FIGURE 54.—Shapes of efficiency curves of boiler No. 1 when platted on the weight of air passing through the flues as abscissæ.

becomes large, on account of the high velocity. Inasmuch as the conductivity of the plate is constant, the temperature difference $(t_1 - t)$ in equation (18) must be increased in order that the increasing quantity of heat can be transmitted through the plate. Since t is approximately the temperature of the boiler water, and therefore can not be lowered, t_1 continues to rise until it nearly equals T , the temperature of the moving gases, when w becomes nearly infinite. Under such a condition the amount of heat absorbed by the boiler is a finite quantity, but as w is nearly infinite the heat available for the boiler is approaching infinity. The true boiler efficiency then becomes a ratio of a finite quantity to an infinite one.

The intrinsic properties of the four equations for E_4 , discussed in the preceding paragraphs, are interesting from a mathematical point of view inasmuch as they show the effect of the various factors.

The equations deserve especial consideration because they are derived in a rational way from fundamental principles in the theory of heat and the theory of gases. To a practical man the equations are interesting, because within the workable range of the rate of heat transfer the value of E_4 obtained from one equation agrees fairly well with E_4 obtained by means of another. Moreover, the efficiencies thus calculated agree with actual results obtained from experiments with laboratory as well as with large boilers. Of course it can be reasonably expected that equation (18), on account of its containing

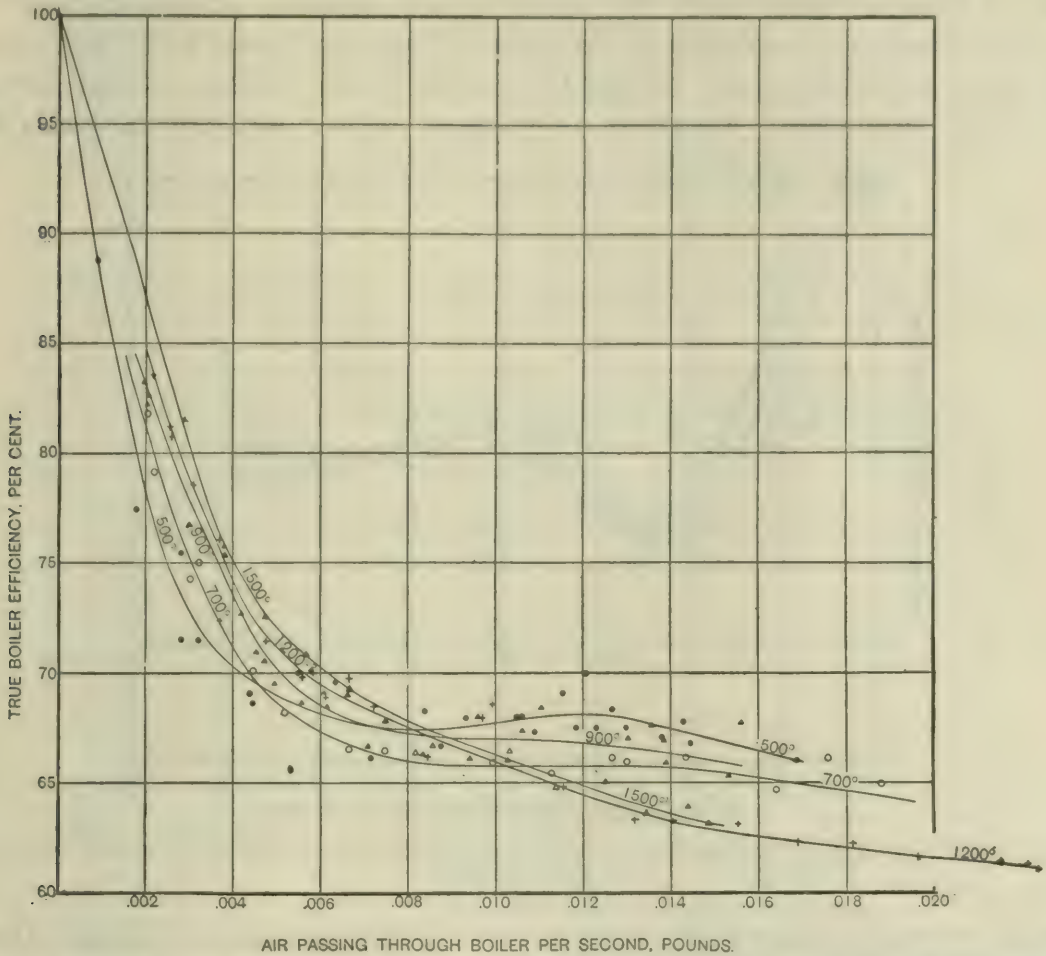


FIGURE 55.—Shapes of efficiency curves of boiler No. 2 when platted on the weight of air passing through the flues as abscissæ.

the largest number of the influential factors, gives values closest to the actual.

The true boiler efficiencies of all the tests made on the small boilers were platted on w , the weight of gases, as abscissæ, and are given in figures 54, 55, 56, and 57. The general shape of the curves in these figures is like that of the graph of equation (37). In figure 56 the sudden rise of the true boiler efficiency after w , the weight of gases, is increased above 0.005 pound per second, can not be explained now. It seems that up to a velocity of about 0.005 pound per second the gas flows in parallel streams, but at a higher velocity violent

eddying starts in the gas stream and increases the rate of heat impartation, thereby raising the efficiency. This factor of sudden eddying is not embodied in any of the equations.

INFLUENCE OF THE CROSS SECTION OF THE GAS STREAM ON THE RATE OF HEAT ABSORPTION BY CONVECTION.

The laboratory experiments have shown that flues of small diameter absorb more heat from each pound of gas passing through them than do flues of large diameter and the same length, although the latter flues contain more effective heating surface. Similarly locomotive boilers, on account of their small flues, are found more efficient as heat absorbers than horizontal multitubular boilers having flues of large diameter. These facts suggest rather strongly a general

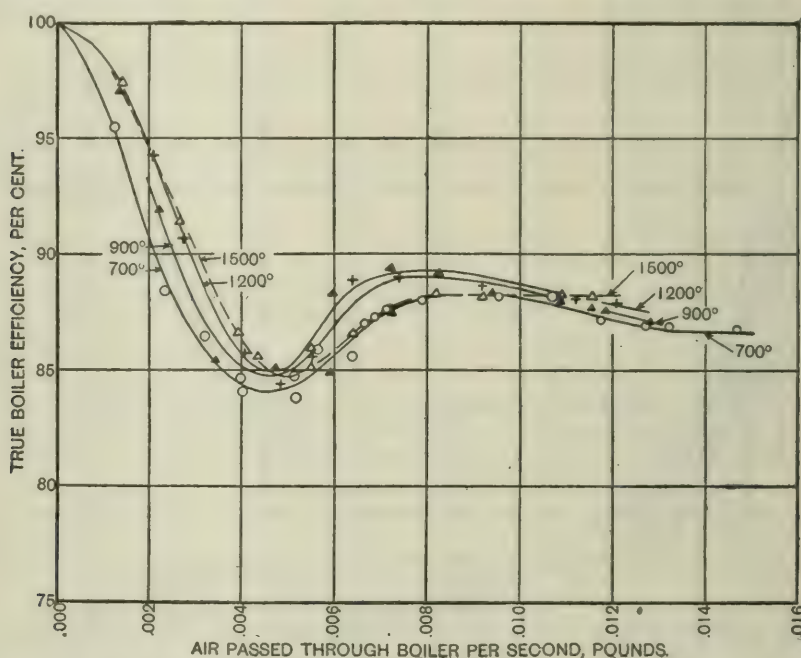


FIGURE 56.—Shapes of efficiency curves of boiler No. 3 when platted on the weight of air passing through the flues as abscissæ.

relation between the cross section of a flowing stream of gas and the effectiveness of a heating surface.

At the beginning of the section on convection it was said that the rate at which heat is imparted to the dry surface depends on the number of gas particles coming in contact with the surface. Obviously, the closer these gas particles are to the surface the sooner they will reach it and the greater the number of contacts each will make in a unit of time. Now, in a gas passage of small cross section the particles of gas are on the average closer to the dry surface than they are in a passage of a larger cross section; therefore the gas particles in the small passage will make proportionately more contacts in a unit of time than the gas particles in the large passage. This principle can be more specifically illustrated by two cylindrical flues

of different diameters. Figure 58 shows a cross section of two flue tubes, one of 4-inch diameter and the other of 2 inch. Let the dots represent particles of hot gases. It will be seen that the shortest distance from any gas particle, A , in the first dot circle of the large tube to the nearest point, a , of dry surface is the same as the shortest distance of a similar dot, A , in the small tube. However, the gas particle does not necessarily move to the nearest point, a , of the dry

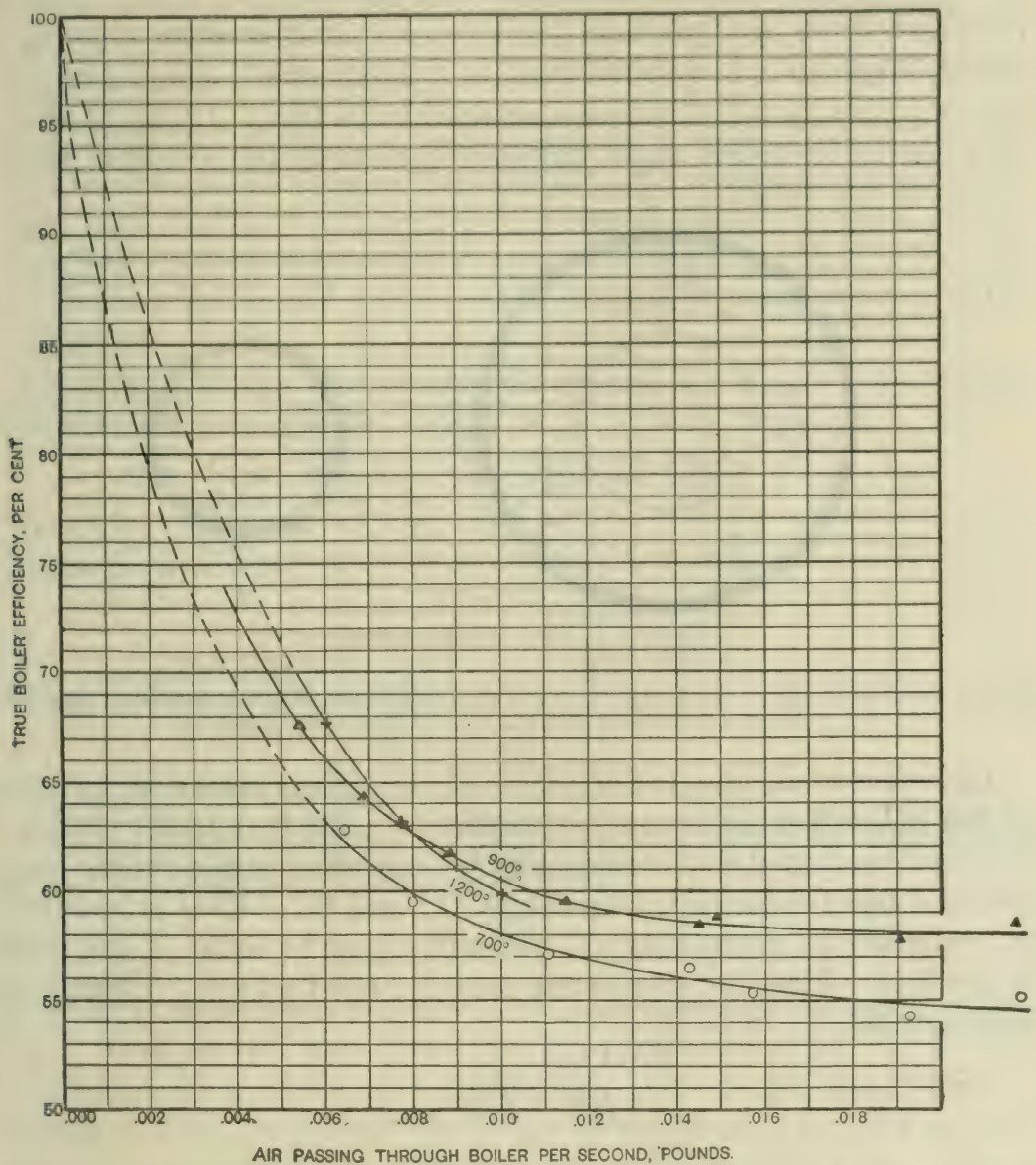


FIGURE 57.—Shapes of efficiency curves of boiler No. 4 when platted on the weight of air passing through the flues as abscissæ.

surface (circumference); it may move to any of the points a_1 , a_2 , or a_n . But any other point of the dry surface is much farther from the gas particle in the large tube than is a point similarly located in the small tube, and therefore the average distance of the gas particle A from the points of the dry surface is greater in the large tube than in the small one. In like manner the average distance of any gas particle from the dry surface in the large tube can be shown to be greater

than the distance of a similarly located gas particle in the small tube. Point A has been used in the argument because at first glance it may appear to be the same distance from the dry surface in both tubes. The fact to be borne in mind is that any gas particle in the flue may come in contact with any of the points in the dry surface. The factor that determines the number of contacts is the mean distance of all the gas particles within the tube from all the points on the dry surface. The shorter this distance the quicker the gas particles make contacts. It hardly needs any mathematical proof that this mean distance is always larger in a large tube than in a small one; a short examination of figure 58 will convince a reader of this fact. It can be shown by calculus that this mean distance is directly proportional to the radius of the tube.

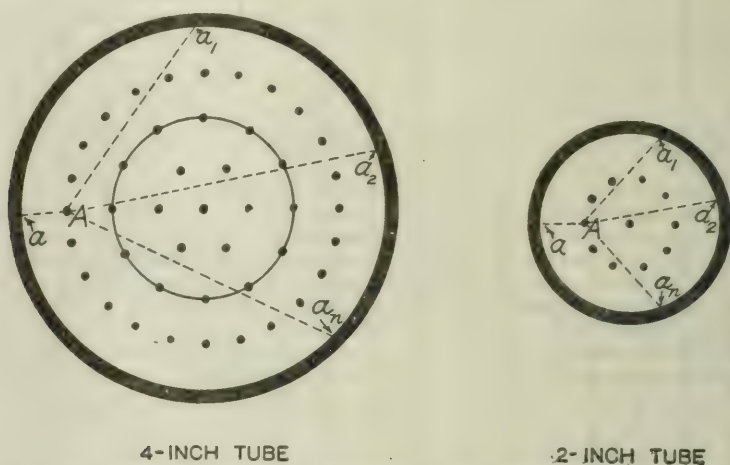


FIGURE 58.—Cross section of two flues showing that the average distance of the particles of gases to the dry surface is shorter in the small flue than it is in the large flue.

The effect of varying the diameter of round flues on the temperature of the gases flowing through them and on the true boiler efficiency can be determined mathematically by introducing the radius factor into the two fundamental equations (1) and (7). This introduction can be made by expressing the velocity of gas in terms of the radius of the flue. First, let us assume that $T - t_1 = T - t = \theta$. Then in the equation (7)

$$H = Cqv\theta,$$

$$v = \frac{C(\theta + K)w}{r^2}, \text{ and}$$

$$q = \frac{C}{(\theta + K)}, \text{ as before.}$$

Substituting the values of v and q in (6) and expressing H in terms of w and θ we have

$$(38) \quad -swd\theta = \frac{Cw\theta}{r^2} \times 2\pi r dx.$$

From this differential equation we obtain in a similar way, as from equation (23), the expression θ_x , the temperature at any point x in the flue, in terms of the initial temperature and the radius of the flue.

(39)

$$\theta_x = \theta_0 e^{-\frac{cx}{r}}$$

Therefore the expression for the true boiler efficiency for varying diameter of flues is

(40)

$$E_4 = 1 - e^{-\frac{cl}{r}}$$

Equation (39) states that with the same length of tubes and the same initial temperature the final temperature drops as the diameter of the flues decreases. Therefore to obtain low flue-gas temperature reduce the diameter of the flues.

Equation (40) states that with the same length of tubes the true boiler efficiency increases as the diameter of the flues decreases. To obtain high efficiency reduce the diameter of flues.

For water-tube boilers we might reason by analogy that to increase the efficiency we should decrease the "hydraulic mean depth" of the gas passages.

A somewhat more complicated expression for E_4 can be obtained when starting from the equation (1):

$$H = A(T - t_1) + Bqv(T - t_1).$$

Assuming again that $(T - t_1) = T - t = \theta$,

Making substitution for q and v as before we have

$$(41) \quad -swd\theta = \left(A\theta + \frac{Bw\theta}{r^2} \right) 2\pi r dx.$$

From equation (23) we get

$$(42) \quad E_4 = 1 - e^{-\frac{Aclr^2 + BKwl}{r}}$$

where cA and BK are constants.

A more correct expression for E_4 could be obtained by introducing into the fundamental equation the relation

$$t_1 = \frac{Hd}{c} + t,$$

but such expressions, on account of their complicated form, would hardly have more value than the two expressions (40) and (42). Equations (39) and (40) have perhaps the greatest practical value on account of their simplicity. They are not as clumsy to work with as they at first appear to be; they can be easily handled when reduced to the following form:

From equations (39) (40)

$$(43) \quad \log_e \theta_l = \log_e \theta_0 - \frac{cl}{r}$$

$$(44) \quad \log_e (1 - E_4) = -\frac{cl}{r}$$

In all equations in this bulletin E must be expressed in fractions of a unit and not in percentage.

COMPARISON OF THEORETICALLY DEDUCED RELATIONS WITH ACTUAL RESULTS.

It may be interesting to know how the actual results obtained from the laboratory experiments with small boilers agree with the simple expression

$$E_4 = 1 - e^{-\frac{cl}{r}}$$

Boilers Nos. 1, 2, and 4 were all of the same length, but the flues were of different diameters; therefore if the values of E_4 which were determined experimentally for each of the boilers, and the values r are substituted in equation (40) cl should remain constant. The values of cl have been calculated and are given in Table 13 in the horizontal line 5. As shown, these values remain very nearly constant for boilers of the same length. For boiler No. 3, which was double the length of the other three boilers, this value is about 25 per cent lower, and for boiler No. 1 it is somewhat higher than the values for boilers Nos. 2 and 4. The true boiler efficiency of boiler No. 3 needs to be raised only about 4 per cent to bring the value of cl to the average. It is to be regretted that the experiments had to be discontinued before tests were made with double-length flues of the larger diameters, 0.230 and 0.293 inches. Results of tests with larger tubes would perhaps be more regular. As shown by the humps in the efficiency curves of boilers Nos. 1 and 3 (figs. 54 and 56), there seems to be a somewhat irregularly varying factor which becomes more prominent with small tubes.

TABLE 13.—Values of cl in equation (40) calculated from results of laboratory experiments with small boilers.

1	Boiler No.....	1	2	4	3
2	Diameter of flues.....inches..	0.175	0.230	0.293	0.175
3	Length of flues, ldo....	8	8	8	16 ^a
4	True boiler efficiency.....per cent..	78	65.5	57	88
5	Value of cl	.265	.245	.247	.186
6	Value of cl				.372

^a This value is 2*l*.

Figure 59 shows graphically the relation between the efficiency and the diameter of the flues as given by equation (40). Each of the curves gives the relation for one constant length of the flues. The solid points denote the results of the experiments with the small boilers; hollow points signify values calculated from the equation. The lower curve is of more significance inasmuch as it has three points experimentally determined. The value of the constant cl used in platting this curve was 0.245. The upper solid curve has

been plotted with the constant cl equal to 0.186, and the dotted line with the same constant as was used for the lowest curve. Attention is called to the fact that although the difference between the two constants employed in calculation was about 25 per cent, nevertheless the one experimental point does not fall very far from the curve of the higher constant.

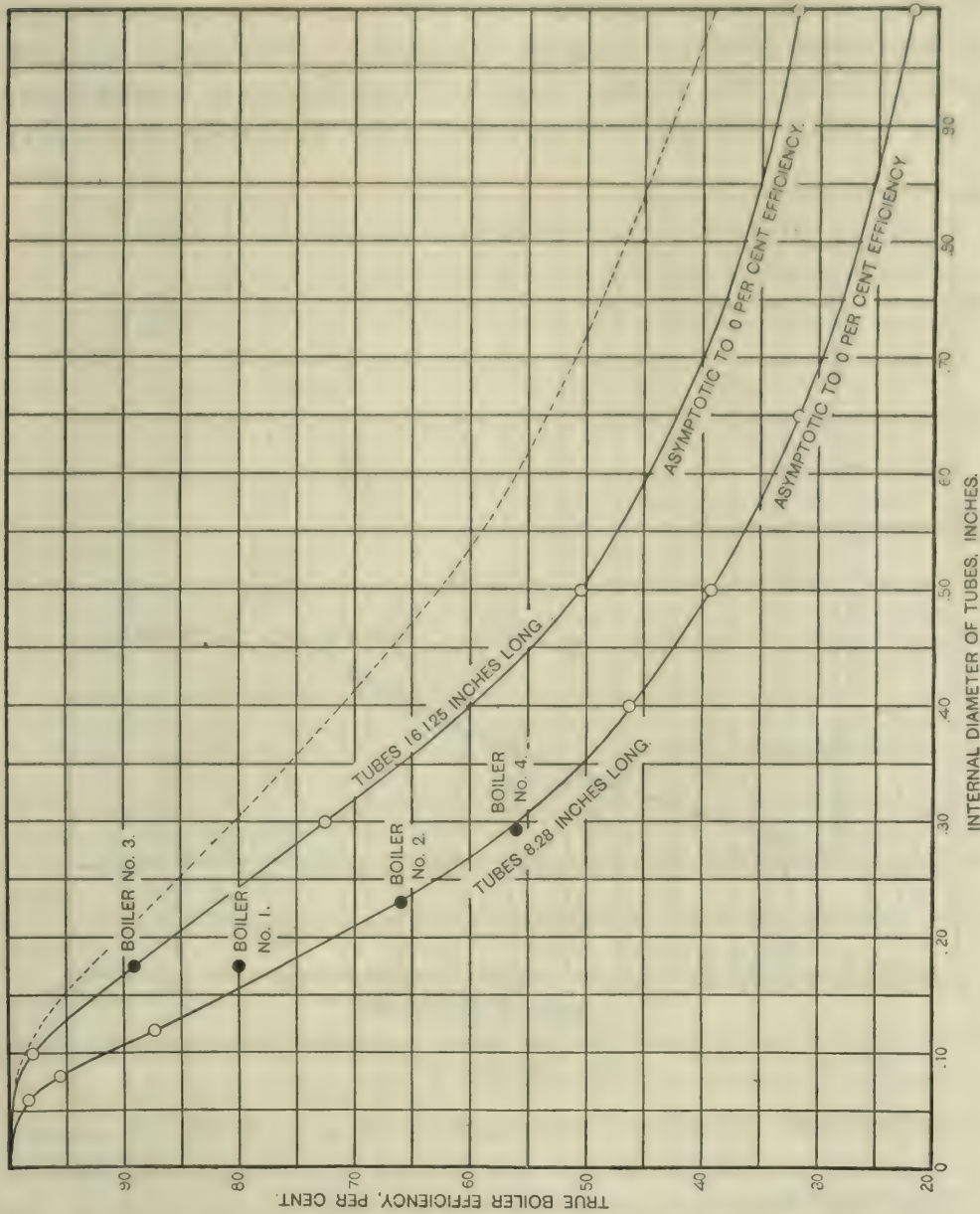


FIGURE 59.—Relation between the true boiler efficiency of the small experimental boilers and the diameter of their tubes when the length of the latter remains constant.

Equation (39) states that the temperature of the escaping gases is directly proportional to the initial temperature, both temperatures being counted above the temperature of the boiler water. This relation means that if the initial temperature is doubled the temperature of the escaping gases is doubled, or if the initial temperature is tripled the final temperature is tripled. To test this approximate

relation the initial and final temperatures of tests made on boilers No. 2 and No. 3 have been platted and are given in figures 60 and 61. Only the curves are shown in the figures, the points having been omitted after the curves were determined, inasmuch as the points fell within a very narrow band. In the figures each curve gives the above relation for one constant pressure drop ("draft") of the gases through the boiler. These pressure drops are indicated with figures, as, for example, 16 IN. P. D., which means a pressure drop of 16 inches of water through the boiler. The arrows point to the curve to which the figures apply. Thus, referring to the figure, when the initial temperature is about 300°F . above the boiler-

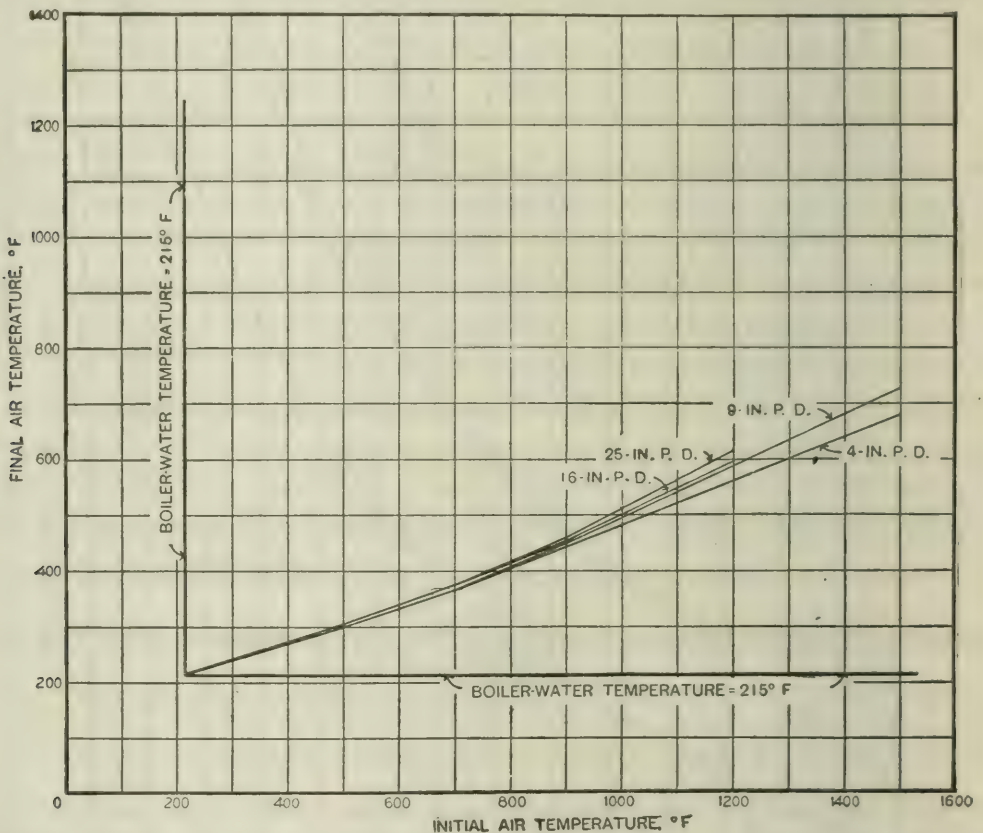


FIGURE 60.—Relation between the temperature of air leaving boiler and the temperature of air entering boiler. Boiler No. 2.

water temperature the final temperature is about 100°F . above that of the boiler water. When the initial temperature is 600°F . above, the final temperature is about 200°F . above that of the boiler water, or, both temperatures are doubled. If the initial temperature is raised to $1,200^{\circ}\text{F}$. above, the final temperature is raised to about 450°F . above that of the boiler water; in this case both temperatures are about quadruples of those in the first case.

It will be noticed that the curves of higher pressure drops are somewhat higher; that is, when more gases are formed through the boiler the final temperature rises a little faster than in proportion to the rise in the initial temperature. This extra rise in the final tem-

perature is known to be due to the rise of temperature of the dry surface of the heating plate due to a higher rate of heat transmission. It has been accounted for in formulas (32), (37), and (42). By selecting the proper constants the graph of equation (32) could be made to agree almost exactly with any curve in figures 60 and 61. It is a question whether the extra trouble in handling the cumbersome equation (32) would be sufficiently rewarded by the small increase in accuracy. A simple equation is far more preferable for practical problems, although it may not be as accurate as a more complicated one. Furthermore, by using fair judgment one can reduce considerably the error due to the inaccuracy of the equation.

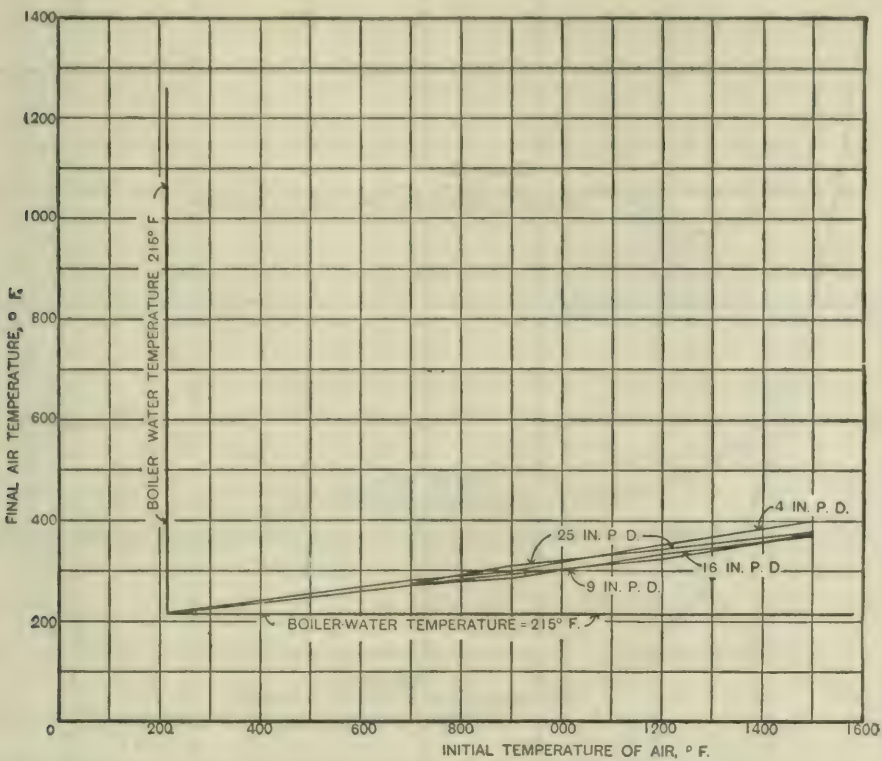


FIGURE 61.—Relation between the temperature of air leaving boiler and the temperature of air entering boiler. Boiler No. 3.

In figure 61 the curves are much nearer straight lines than the curves of figure 60, showing that the simple equation (39) holds much better for longer tubes.

Figures 62 and 63 show how the true boiler efficiencies of the three boilers having flues of constant length but of different diameters vary with the weight of gas passing through the boilers and with the pressure drops through the boilers.

The curves show (a) that the efficiency is higher with tubes of smaller internal diameter; and (b) that beyond certain values of the pressure drops or the weight of gas passing through the boiler the efficiency is nearly constant. The simple equation (40) is applicable

only to this horizontal portion of the curves. For the portion of the curves before they become nearly horizontal equation (42) should be used. One may say, then, that for high rates of working the simple equation is fairly accurate, but that it can not be used when the gases move slowly and the boiler is working at low capacity.

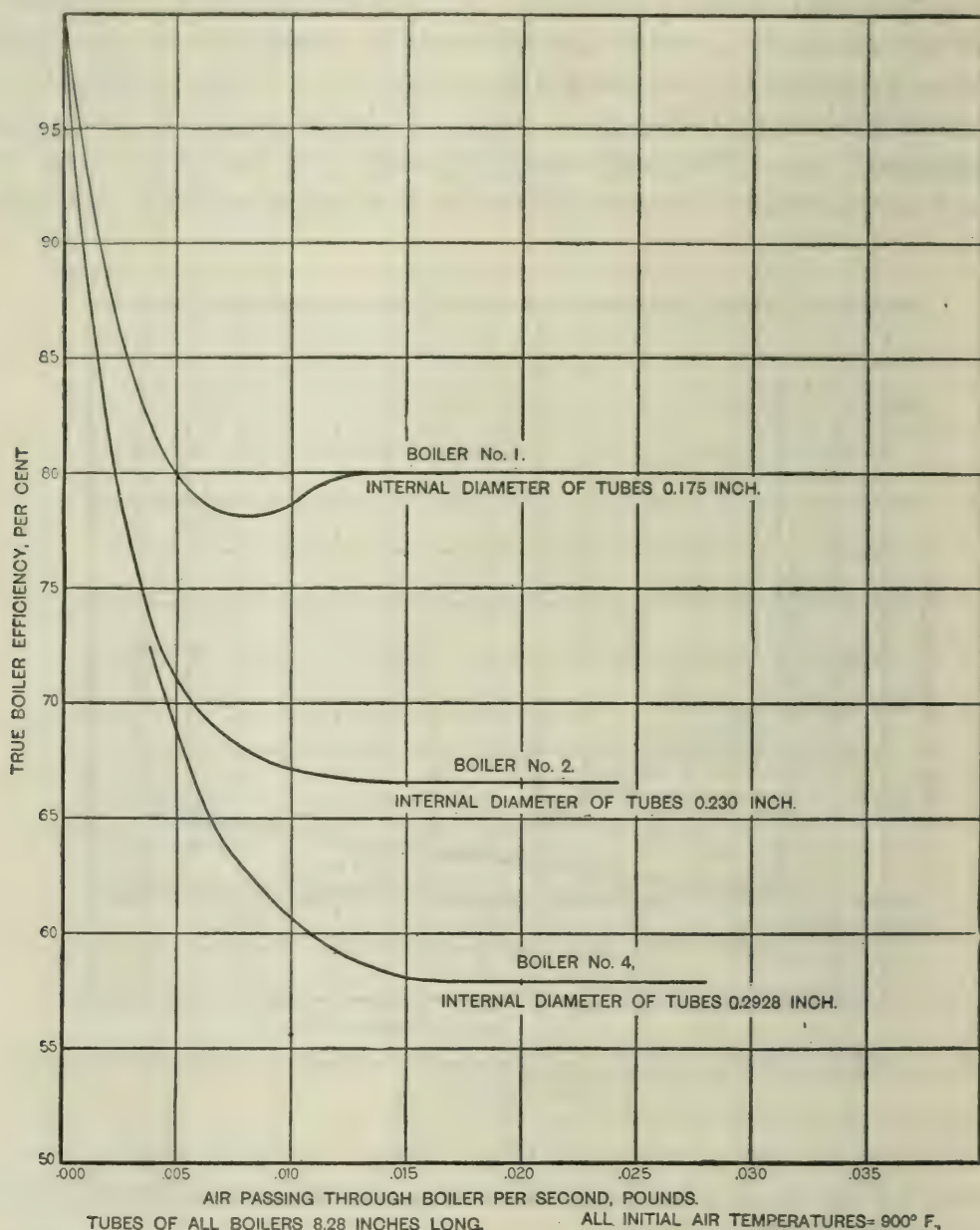


FIGURE 62.—Effect of diameter of flues on the shape and the location of the true boiler efficiency curves. Curves platted on the weight of air passing through boiler as abscissæ.

On the whole, the agreement between the results obtained from the experiments and the relation derived by mathematical reasoning is good. It shows that the direction taken by Prof. Reynolds and followed by Prof. Perry in the rational study of the problem of heat impartation to the heating plates of a boiler by convection is right and that the principles developed by these two pioneers are indeed well worthy of further study.

THE ABILITY OF BOILER WATER TO ABSTRACT HEAT FROM THE WET SURFACE.

It has been explained that the heat reaches the dry surface by convection and by radiation and that it is then transmitted through the plate to the wet surface purely by conduction. The laws which govern these modes of heat transmission have been discussed in their

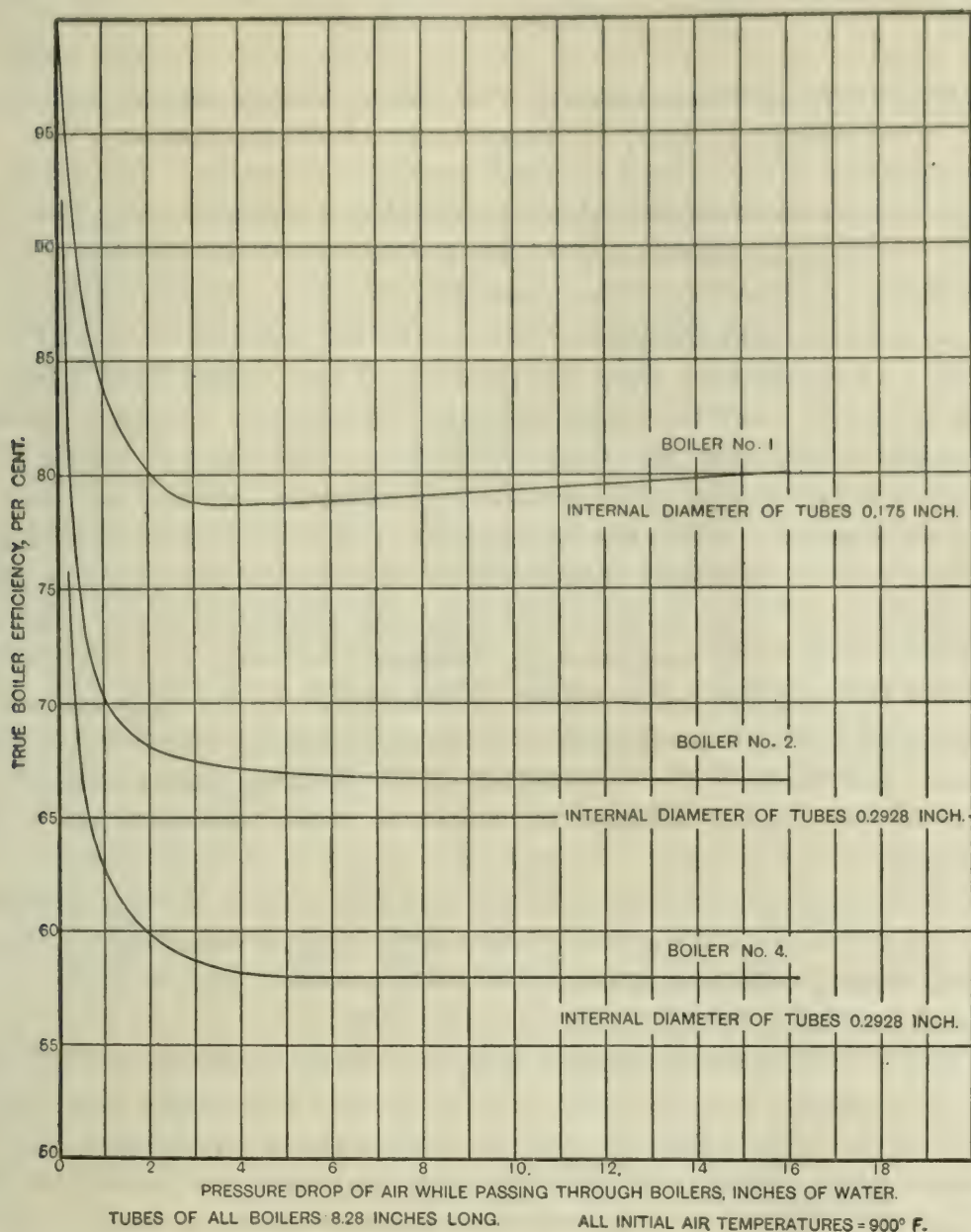


FIGURE 63.—Effect of diameter of flues on the shape and the location of the true boiler efficiency curves. Curves platted on the pressure drop of air through boiler as abscissæ.

special application to each part of the path of the heat travel. It has been stated in a general way that the boiler water abstracts the heat from the wet surface almost entirely by convection. The general law of convection discussed in the preceding chapter is applicable to this part of the path traveled by the heat.

The wet surface of the heating plate has been defined as a thin region in the film of water, or perhaps water and steam, next to the metal, where the heat ceases to travel by conduction and starts to travel by convection. When the heat enters this region perhaps it may raise the temperature of the water somewhat, but mostly it is absorbed by the change of the water into steam. This steam is then torn away from the metal and replaced by water. Thus the heat enters the film of water containing the wet surface by conduction and is taken from it by convection; that is, the particles of water entering the wet surface come in contact with the metal and absorb heat from it by conduction; then, after changing into bubbles of steam, they are removed and are replaced by fresh particles of water. Inasmuch as the exchange of steam bubbles for particles of water is brought about by the current or circulation of the boiler water, the process of heat transfer is convection. The particles of water may be likened to empty buckets and the steam bubbles to full buckets of a heat conveyor. It can be seen that the quantity of heat taken from a unit of area of the wet surface depends upon the number of steam bubbles leaving it; the faster the steam bubbles are taken out of the film the higher will be the rate of heat exchange between the wet surface and the boiler water. Since the absorption of heat by the water changes the latter into steam and does not raise its temperature—that is, since the addition of heat is manifested by the change of state—there is probably little or no temperature difference between the wet surface and the body of the boiler water. On account of the high density of water and its high latent heat, its heat-absorbing capacity is very great. Therefore in the wet surface of the heating plate the displacement of the steam bubbles by water is much slower than is the exchange of the hot and cold particles of gas in the dry surface. In this latter case the heat is sensible, and not only is the specific heat of the gas low, but the gas itself is so light that a large volume must make contact with a unit area of dry surface to impart to it the same quantity of heat as the water can abstract.

Thus, at the average furnace temperature of $1,750^{\circ}\text{F}$. the density of furnace gases is only about one thirty-two hundredth that of the water in the boiler, and the specific heat is about one-fourth. If the average gas particle is reduced by each contact with the unit of dry surface by $1,000^{\circ}\text{F}$., the relative amount of heat given up is

$$\frac{1}{3,200} \times \frac{1}{4} \times 1,000 = \frac{1}{12.8}.$$

A small unit volume of water is able by virtue of its latent heat to absorb relatively about $900 \times 12.8 = 11,520$ times as much heat as an equal volume of gas is able to impart through a reduction in temperature of $1,000^{\circ}\text{F}$. Even if the effective wet surface be reduced by the steam bubbles to one-half, the water has a considerable advantage above the gas. This high concentration of

heat in a small volume of water resembles the high density in the general law of convection for fluids.

The preceding figures justify the statement that the boiler water will absorb from the wet surface all the heat that the gases can impart to the dry surface. The natural water circulation in many boilers is undoubtedly sufficient to replace by water the steam bubbles formed in the wet surface, even though the rate of their formation were several times what it is at present boiler capacities.

ADVANTAGES GAINED BY REDUCING THE TEMPERATURE OF THE GASES BEFORE THEY PASS THROUGH THE BOILER.

It has been sufficiently proven that the temperature of the gases leaving a boiler varies almost directly as the temperature of the gases entering the boiler, both temperatures being reckoned above that of the boiler water; the lower the entering temperature the lower the leaving temperature. It is obvious that if two boilers have exactly similar construction and baffling, the gases in the one which is so set that it receives more heat by radiation from the furnace will have the lower initial and final temperatures, and the efficiency of this boiler will be the higher. The gases themselves radiate but little heat to the boiler, but part of the heat generated by the combustion of the fuel is radiated by the hot fuel bed, the flames, and the furnace directly to the boiler and is never absorbed by the gases. Therefore the latter enter the boiler at a lower temperature than they would if they absorbed all the heat generated in the furnace. This means that any heat which the boiler gets directly by radiation is a clear gain. Of course the presumption is that the combustion and air supply is the same in all cases.

Specific illustration can be made with two similar boilers of the standard Heine type of baffling and setting. Assume that in one of the boilers the first baffle is formed by tiles which entirely encircle the lowest row of tubes, whereas in the other the baffle is formed by flat tiles lying on top of the lowest row of tubes so that the tubes are exposed to direct radiation from the fuel bed and flames. In the first setting the boiler is screened by the tile roof and can not, therefore, cool the fuel bed and the luminous flames so readily as the second boiler. Consequently the temperature of the furnace and that of the gases entering the tube chamber will be lower in the second boiler than in the first one, because the former has already absorbed part of the heat from the furnace by radiation. The result will be that the second boiler will have lower flue-gas temperature and will show higher efficiency.

The engineering experiment station of the University of Illinois made two series of tests on the same boiler. One series was made with the boiler having a tile roof, the first setting described above;

the second series was made with the tile roof removed; that is, the first baffle was formed as in the second setting described above in the second case. The second series of tests showed flue-gas temperatures considerably lower and the over-all efficiency from 3 to 5 per cent higher, demonstrating that the absence of the tile roof allowed the boiler to absorb a considerable percentage of the total heat by radiation from the fuel bed, furnace walls, and flames. The furnace of this particular boiler was equipped with a chain grate stoker. On these tests very little smoke was made with either style of the first baffle. The absorption of heat from the fuel bed and the flames, if carried to an extreme, might result in incomplete combustion and smoke.

SUMMARY OF PRINCIPAL DEDUCTIONS.

When fuel burns in a boiler furnace most of the heat generated is immediately absorbed by the gases that result from combustion. These gases then pass over the heating surface of the boiler and impart the heat they contain to the boiler; this heat impartation takes place almost entirely by convection. The gases then act as a conveyor of heat; that is, they carry the heat from the burning fuel to the "dry surface" of the boiler plate. Therefore the amount of heat a boiler receives per unit of time depends to a great extent on the utilization of the heat-conveying or convecting properties of the gases.

The conductivity of the metal forming the heating plate, even though the plate has a moderate amount of coating, is so high that any quantity of heat that can be imparted to the dry surface by convection can be transmitted through the heating plate without difficulty. Water has such high heat-abstracting ability that it can absorb heat as fast as the heating plate is able to conduct it. The slowness of the process of heat transmission into a boiler is due to the slowness with which heat, in present practice, is imparted to the dry surface.

It has been shown that the rate at which the gases transfer heat to the dry surface of the boiler depends on the following factors:

(a) On the difference of temperature between the gas and the dry surface.

(b) On the density of the gas.

(c) On the velocity at which the gas flows over the dry surface.

As to the first factor it must be said that boiler furnaces always should be operated at the most economical temperature. Generally this is not the highest practicable temperature, as the necessary reduction of the air supply may cause part of the combustible to escape unburned from the furnace; further, a very high temperature may not be desirable on account of expensive repairs on the fire-brick lining in the furnace. Considering these two limitations the temperature should be as high as feasible in order to make the quantity of heat

available as large as possible. In well-managed plants the furnaces are operated at the best economical temperature and therefore there is not much chance of improvement by increasing the first factor.

The density depends on the absolute temperature of the gases and is therefore fixed when the temperature is determined.

The velocity of gaseous products of combustion is independent of their temperature and their density and therefore can be increased at will whenever a higher rate of steam production is desired. It is this velocity factor which enables locomotive and marine boilers to produce steam at two or three times the rate customary with stationary boilers. In the first two types of boilers double or triple capacity is obtained by forcing over their heating surface two or three times the weight of gases at two or three times the velocity. The same results can be effected in stationary boilers; their capacity can be increased several times by forcing several times the weight of gases over their heating surface at several times the velocity.

It has also been shown, both by the results of experiments and by theoretical considerations, that more heat can be abstracted from the same weight of gas if the length of the gas passage is increased or the cross section of the passage is reduced; the second condition must always include the reduction of the "hydraulic mean depth" of the gas stream. Both of these conditions increase the number of contacts the particles of gas make with the dry surface. With any given velocity, lengthening the gas path lengthens the time during which each particle can make contacts with the dry surface; reducing the cross section or the "hydraulic mean depth" reduces the mean distance of each particle of gas from the dry surface of the plate so that the particles of gas can reach the surface in less time, and in the same available length of time make more contacts. Abstracting more heat from the gases means higher boiler efficiency. The length and the cross section of a gas path is really the arrangement of the heating surface with respect to the gas stream, so that one may say that the efficiency of the boiler depends on the arrangement of the heating surface with respect to the flow of gases.

The preceding deductions can be summarized in the following brief statements:

The capacity of a boiler can be increased by forcing a greater weight of gases through the boiler.

The efficiency of a boiler as a heat absorber can be increased by arranging the heating surfaces in such a way that the gas passages are long and of small cross section, so that they have a small "hydraulic mean depth."

PART V.

APPLICATION OF THE LAWS OF HEAT TRANSMISSION TO STEAM PLANTS AND STEAM-BOILER AND STEAM-PLANT DESIGN AND TO STEAM BOILERS ALREADY INSTALLED.

APPLICATION OF LAWS TO STEAM-BOILER AND STEAM-PLANT DESIGN.

The application of the principles of heat transmission offers an excellent opportunity for economic improvements in the steam-boiler as well as in steam-plant design. It seems feasible to increase the capacity of boilers several times; and at the same time, by properly arranging the heating surfaces in the boilers, to raise the efficiency perhaps several per cent. Increasing the capacity of boilers would reduce the first cost of the installation, and consequently less interest would have to be paid on a smaller investment. Increasing the efficiency would reduce the coal bill. Under such conditions power could be produced much more cheaply than it is at present.

INCREASING CAPACITY.

Considering the high heat conductivity of metals, there is no reason why boilers could not be made to generate steam at three or four times the rate they do at present by simply forcing about three or four times the weight of gases over their heating surfaces. This may be considered a conservative statement, as much higher rates are possible and have been attained.

It may at first seem that the power required to force such large quantities of gas through the boilers would be so great that the cost of installing and operating the "draft" appliance would offset any gain from the increased capacity of the boiler. However, such is not the case. It is true that the power required to push the gases through the boiler increases as the cube of the capacity the boiler develops, so that to triple the capacity of a boiler the power required to push the gases would have to be increased 3^3 , or 27 fold; or, to quadruple the capacity the fan power would have to be 4^3 , or 64 times as large as for single capacity. The one feature that makes high capacity ^a possible is the fact that the power really consumed in ordi-

^a Ray, W. T., and Kreisinger, Henry, The significance of drafts in steam-boiler practice. Bulletin 21, Bureau of Mines. 62 pp

nary cases in pushing the gases through the boiler and furnace is so small that even after it is multiplied by 64 it still remains only a small fraction of that developed in the boiler. Distrust in the possibility of working steam boilers harder rests on past experience. Engineers are accustomed to hear that the steam consumption in mechanical-draft appliances is from 1 to 2 per cent of the total steam generated by the boilers which are served by the appliances. With such extravagant "draft" productions, of course large increase of boiler capacity would not be practicable. Thus, suppose that a fan producing "draft" for a boiler battery at ordinary rates of steaming consumes 2 per cent of the steam generated in the battery. If the capacity is to be quadrupled, the fan would apparently have to do 4³, or 64 times, as much work as before. Since 4 times the capacity is developed, the steam consumed would be $\frac{64 \times 2}{4} = 32$ per cent of the total steam generated at the quadrupled capacity. This, of course, is a considerable percentage, high enough to make any commercial consideration out of question.

However, a little more detailed investigation will show that with present mechanical-draft outfits about 90 per cent of the energy that can be developed from the steam consumed in the fan engine is wasted in the crude inefficient engine, in the still cruder fan, and in the long leaky gas or air ducts having many sharp turns; and that only about 10 per cent is actually expended in pushing the gases through the furnace and boiler. The power actually needed to force the gas through the boiler can be easily figured from the volume of gas to be displaced per minute and the pressure against which the gas must be moved. This power in foot-pounds is equal to the product of the pressure in pounds per square foot and the cubic feet of air displaced; the following expression gives it in horsepower:

$$\text{Horsepower} = \frac{\text{pressure (lbs. per sq. ft.)} \times \text{cu. ft. gas displaced per minute}}{33,000}$$

In specific cases the power expended in pushing gases from the ash pit to the base of the stack or to the uptake can be figured as shown in the following illustration:

Assume that in a battery of boilers ordinarily developing 1,000 boiler horsepower the equivalent evaporation is 9 pounds of water per pound of dry coal of average quality, and that it takes 15 pounds of air to burn 1 pound of dry coal; then the figures are:

$$\text{Weight of coal burned per minute} = \frac{1,000 \times 34.5}{9 \times 60} = 63.9 \text{ pounds.}$$

$$\text{Weight of air used per minute} = 63.9 \times 15 = 958.5 \text{ pounds.}$$

$$\text{Volume of air per minute} = 958.5 \times 13 = 12,460 \text{ cubic feet.}$$

Ordinarily the pressure drop from ash pit to uptake seldom exceeds 0.75 inch of water, but for convenience let the pressure drop in this case be taken as 1 inch.

$$\text{One inch of water} = \frac{62}{12} = 5.16 \text{ pounds per square foot.}$$

$$\text{Horsepower} = \frac{12,460 \times 5.16}{33,000} = 1.95.$$

Now, according to the cube law, if the capacity of this battery is to be quadrupled the power needed to move the gases would be 4^3 , or 64 times 1.95, which is approximately 125 horsepower.

Surely in a unit of this size one engine brake horsepower ought to be produced with one boiler horsepower, and fans having an efficiency of 75 per cent can be built. With such outfits the consumption of steam for the quadruple capacity would then be

$$\frac{125}{4,000 \times .75} = 4.2 \text{ per cent of the total steam generated.}$$

When one considers that in big power stations power is produced in large electrical units at about half the steam consumption assumed above, and also remembers that the fans could be driven electrically, the quadruple capacity appears commercially very feasible—especially if by proper arrangement of the heating surfaces the efficiency of the boilers is increased.

If one considers the sizes of fans usually seen in power plants at present, he may perhaps make the objection that if the capacity of boilers were quadrupled the fans would be so large that they might take up the greater part of the space saved by making the boilers do more work. There is, however, no substantial reason for fearing such conditions. The fans used at present are unduly large because of their low efficiency; in different constructions the efficiency ranges from 80 down to 10 per cent, and in most is perhaps much closer to 10 than to 80 per cent. Fast-running fans of large capacity and high efficiency are now being put on the market. Such fans could be easily placed under or above the boiler they were to serve without being conspicuous on account of their size.

As an example of the feasibility of increased boiler capacity, the reader is referred to United States Geological Survey Bulletin No. 403^a, wherein are described 21 tests made on a water-tube boiler of the United States torpedo boat *Biddle*. This particular boiler had a total heating surface of 2,776 square feet. It would, therefore, be rated in land practice as a 277.8-horsepower boiler. On some of the tests as much as 915 boiler horsepower were developed—about 3.3 times its rating in land practice—and still higher capacity was

^a Reprinted as Bulletin 33, Bureau of Mines.

possible. A blowing fan forced the air directly into the fire room; the latter was as nearly air-tight as practicable and was kept under pressure. The wheel of the fan was 5 feet in diameter and 14 inches wide. It was placed directly on the fire-room wall and had no casing, but was protected by a wire screen. The blades were curved in the direction opposite to the rotation of the wheel. At the high boiler capacity mentioned the speed of the fan was about 850 revolutions per minute, and the power required was about 15 horsepower.

The question may be asked, Why are the fans used at present so crude and inefficient? The answer is that the manufacturer had to sell them cheaply in order to sell them at all, and cheapness and good design and efficiency in apparatus do not go together. Really, there was hardly anything to be gained by more refined and efficient mechanical-draft outfits. Suppose that the steam consumption would by refined apparatus be reduced from 2 per cent to $\frac{1}{2}$ per cent of the steam generated in the boiler, would any plant manager be willing to pay, say, double the price for the installation of a more efficient mechanical-draft apparatus to save at most $1\frac{1}{2}$ per cent of the steam?

Another objection that can be made to forcing steam boilers to high capacity is the alleged higher moisture in the steam. Here again one can refer to locomotive and torpedo-boat boilers; these types of boilers are working at about three times the rate customary in stationary boilers and work under conditions favorable to priming; still there is no particular trouble from moisture in steam. At large steam-turbine plants, where steam is highly superheated, complaints of higher moisture in steam are not heard.

In designing a steam-generating apparatus to produce steam at high rates, probably the greatest difficulties will be met in the furnace. To produce four times the weight of hot gas, four times the weight of coal must be burned in the same time. To burn coal thus rapidly without large losses through incomplete combustion of gases and tar vapors, driven off from the soft coals upon heating, and losses in sparks—losses which always accompany the high capacity of the locomotive and torpedo-boat boiler—the furnace would have to be the larger part of the steaming apparatus.

To prevent the escape of unburned gases and tar vapors, the principle of slow heating^a should be used in stoking the coal, in order to avoid the distillation of heavy hydrocarbons which are difficult to burn. Combustion space should be provided in which to burn the lighter and more quickly burning compounds.

In the locomotive and torpedo-boat boiler the grates are comparatively small, so that in order to force a large quantity of gases through the fuel bed the velocity of the gases must be high. This

^a Bulletin 33, Bureau of Mines, p. 44.

high velocity lifts small particles of burning coal from the fuel bed and carries them through the boiler. To avoid the loss from sparks in stationary boilers, the velocity of the gases through the fuel bed should be kept so low as not to start the sparks on their way to the stack. This end can be attained by increasing the grate area. The increase in the grate area will also reduce the resistance to the flow of the gases and thereby reduce the power needed to push the gases through the fuel bed. The velocity through the boiler itself should be high, so as to carry through the boiler any solid particles that might settle on the heating surface. If the gases flow faster through the boiler than through the fuel bed, it is obvious that any solid substance that has been light enough to be lifted from the fuel bed will surely be carried through the boiler by the higher velocity. The work of H. G. Stott and W. S. Finlay, jr., discussed in subsequent pages (pp. 147-155) of this bulletin, shows a good method of increasing the grate area. To burn four times the weight of coal on double the grate area would require much less pressure drop through the fuel bed than if it had to be done on the present small grates. Increasing the grate area would reduce the fan work considerably.

From the foregoing discussions it can be seen that the obstacles in the way of higher boiler capacities are superficial, and if the great advantage in the reduction of the first cost of a boiler plant is considered, it seems that the high-capacity boiler is bound to come in the near future. This reduction in boiler plant will give steam power a greater advantage over gas power in large central stations.

GAS AND STEAM POWER AS COMPETITORS IN LARGE CENTRAL STATIONS.

A few years ago the gas engine apparently threatened to displace the steam engine, and even the steam turbine, wherever mechanically feasible; that is, wherever real estate was cheap. It has not done so, however, and if one seeks the reason why it has not he has only to ponder some figures in a paper by William L. Abbott, chief operating engineer of the Commonwealth-Edison Company of Chicago.^a

Mr. Abbott states that of the total operating revenue received by large central-station companies, 50 per cent goes out for "fixed charges," consisting of interest or dividends on investment, sinking fund, or depreciation, taxes, and insurance; 30 per cent for operating expenses of all kinds whatsoever, and 20 per cent for general expenses, including salaries of general officers, advertising, rentals, etc. Of the 30 per cent spent for operating expenses, 20 per cent goes for coal; in other words, only 6 per cent of the company's gross operating income goes for fuel.

^a *Electrical World*, vol. 54, October 28, 1909, pp. 1025-26.

Mr. Abbott specifically states that the fuel saving attained by a gas-engine plant would be more than overcome by the heavier fixed charges on such a plant; especially would this hold if the heavier labor costs in gas-engine plants using producers are taken into consideration. Now, it is easy to see that a gas-engine plant that would save half the coal (an improbable attainment) would save only 3 per cent of the gross operating revenue, and that the increased labor costs alone might approach the saving, not to consider the added bond interest.

The following figures of the cost of installation of steam-turbine plants and of producer-gas engine plants of large capacities^a are taken from a chart compiled by Edwin D. Dreyfus.

Cost of installation of steam-turbine and of producer-gas engine plants.

Total plant capacity installed.	Cost of steam-turbine plant per kilowatt.	Cost of producer-gas engine plant per kilowatt.
<i>Kilowatts.</i>	<i>Dollars.</i>	<i>Dollars.</i>
4,000	75	110.00
8,000	65	104.00
12,000	58	102.00
16,000	53	100.00
20,000	49	99.00
24,000	47	98.50
28,000	46	98.30

The cost of labor Dreyfus gives as 19 to 13 per cent of the total power-plant charge in the steam plant and from 23 to 20 per cent in the gas-producer plant.

The above status of the comparison is present day. It has been repeatedly pointed out in this bulletin that there are bright prospects of getting two or three and perhaps four times the amount of steam from a given boiler-house investment at a higher efficiency than is usual at present. If steam power-house builders should put as much extra money into air preheaters, fans, economizers, etc., as the advocates of gas engines desire put into gas installations, the additional equipment would work some surprises; whether the greater investment would be sound commercial economy in most cases the authors of this bulletin do not undertake to say.

It is well known that in modern turbine power plants the boiler room is the costly portion; if it be assumed (probably safely for our purpose) that only one-fourth of an electricity-supply company's money is in its power houses, and that three-fifths of this latter investment is in the boiler rooms, then to reduce the boiler-room investment only one-half would reduce the total bond interest 7.5 per cent, a liberal dividend on the average amount of capital stock—and it

^a "Prime movers for central stations, their economic relation," a paper read by Edwin D. Dreyfus before the Association of Iron and Steel Electrical Engineers, September 28, 1911.

must be borne in mind that such companies always will be run for the benefit of the stockholders.

The reader will surmise by this time that the imminence of new competition was the best thing that ever happened to steam engineering; it is altogether possible that the rapid strides now being made in boiler-turbine plants will keep them commercially in advance of gas installations, so far as one can now see, except in those cases where gas is to be had for next to nothing or as a by-product.

INCREASING EFFICIENCY.

By applying the principles of heat transmission developed in the general discussion it would be possible to design with fair accuracy a boiler for any desired efficiency. It has been shown that the efficiency of a boiler can be increased by increasing the length of gas path or by reducing the cross section of the gas passages. By the latter is meant the cross section of each elementary stream and not the sum of all of them, as for example in the locomotive boiler it is the cross section or the size of the tubes which determines the efficiency and not the number of tubes. In other words, the reduction of cross section must reduce the "hydraulic mean depth" if the efficiency is to be increased. The "hydraulic mean depth" is the quotient of the area of the cross section of the gas stream divided by the perimeter formed by the heating surface of the boiler and touched by the gases. Thus, if the size of the tubes in fire-tube boilers is reduced, the area is decreased in proportion to the square of the diameter of the tubes, whereas the perimeter is reduced only in proportion to the first power of the diameter; therefore the quotient, or the hydraulic mean depth decreases in proportion to the diameter of the tube. If, however, the size of tubes is kept the same but their number is reduced, then the area decreases at the same rate as the perimeter, and therefore the "hydraulic mean depth" remains constant.

In a similar way the effective reduction in the cross section of the gas stream in a water-tube boiler is attained only by placing the water tubes closer together and not by reducing the number of tubes or baffling the boiler. So that to be effective a reduction in cross section of the gas stream must always be accompanied by a reduction of "hydraulic mean depth," or the mean distance of the particles of gas from the dry surface. The last expression will perhaps appeal best to the reader inasmuch as the reasoning back of it is a little plainer—the closer the gas particles are to the dry surface the more contacts they can make with it in a unit of time. The reasoning for a long gas path is that the longer the path the longer the time for the gas to pass over the heating plate and the more contacts the gas particles make with the dry surface.

It has been shown that for fire-tube boilers the relation between the true boiler efficiency and the length and diameter of the tubes can be approximately expressed by the equation

$$E = 1 - e^{-\frac{cl}{r}}.$$

The relations can be shown better by the graphical representation of this equation given in figures 64 and 65. These two charts

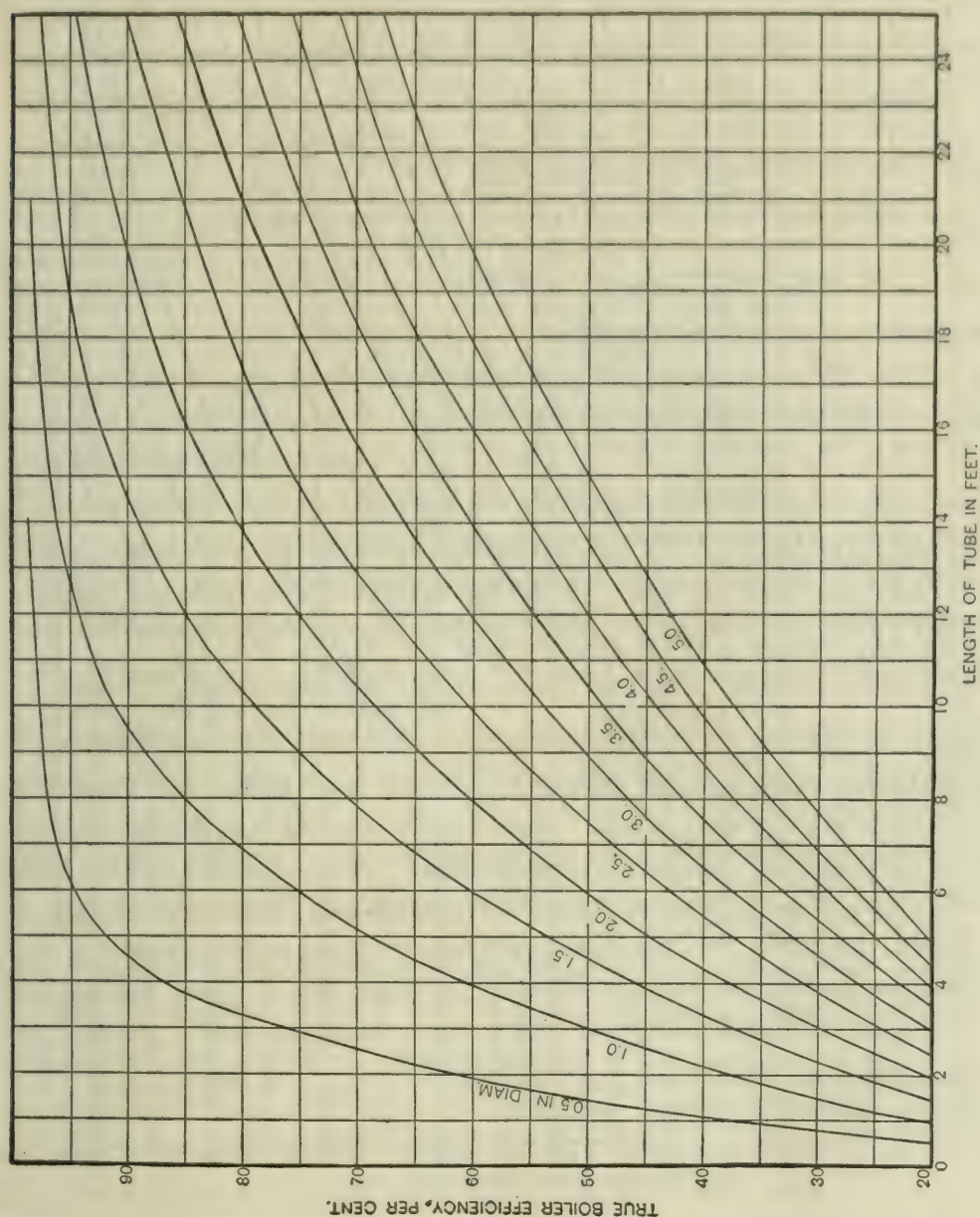


FIGURE 64.—Relation between the true boiler efficiency and the length of tubes.

enable anyone to solve for the unknown value when the other two are given, without complicated calculation. Both figures are based on the dimensions and efficiency of the locomotive boiler tested by the United States Geological Survey at the Seaboard Air Line shops, Portsmouth, Va.^a The tubes of this boiler were 2 inches in diameter

^a For complete report of these tests, see Bulletin 34, Bureau of Mines.

by 14 feet in length, and the true boiler efficiency was approximately 80 per cent. This one set of values is given in the two figures by the solid small circle. Other values were figured and platted by means of equation (40). The values obtained from these figures are therefore more particularly adaptable to locomotive boilers.

Each curve in figure 64 shows how the efficiency varies with the length of the tubes when the diameter remains at the constant

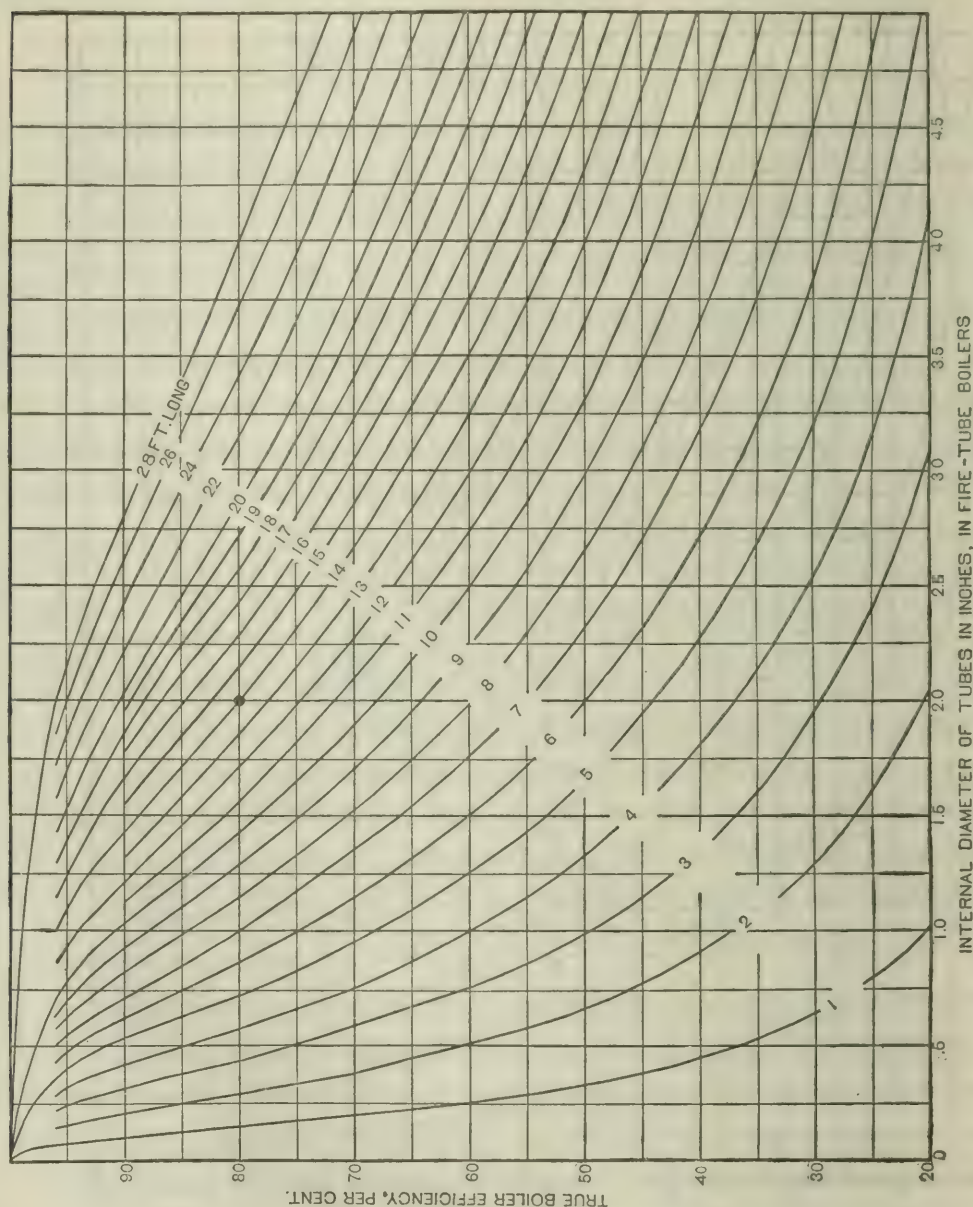


FIGURE 65.—Relation between the true boiler efficiency and the diameter of flues.

value indicated on each curve. Thus, on the curve of 2-inch diameter when the length, given at the base of the figure, is 14, the true boiler efficiency is 80 per cent, the starting point. Now, if one wishes to know how long tubes of the same diameter would have to be to give a true boiler efficiency of 90 per cent, he should follow the curve of 2-inch diameter until he reaches the horizontal line of 90 per cent, and from the point of intersection drop a vertical line to

the scale at the foot of the chart; this scale shows that the length must be 20 feet.

Most horizontal multitubular boilers have tubes 3.5 inches in diameter and about 16 feet long; what is their true boiler efficiency? Follow the curve of 3.5-inch diameter until the vertical line starting from the point of 16-foot length is reached; then from the intersection follow the horizontal line to the efficiency scale on the left. This scale indicates the efficiency of the multitubular boiler to be about 65 per cent.

Suppose a boiler designer wishes to design a boiler that will have a true boiler efficiency of 95 per cent, and wants to use tubes of 1 inch diameter, how long will the tubes have to be? Follow the curve of 1 inch diameter until the efficiency line of 95 per cent is reached; the imaginary vertical line indicates that the length of the tubes must be about 12.75 feet.

By following any constant efficiency line one finds that as the diameter is doubled the length is doubled, so that it is possible to say that tubes having a constant ratio of diameter to length have also constant efficiency, no matter what the diameter or the length or the heating surface may be.

The curves of figure 65 give the relation between the true boiler efficiency and the diameter of tubes for constant lengths indicated on each curve. This chart can be used to advantage to find any one of the three factors when the other two are given, but it was particularly designed for finding the diameter of the tubes required for any given length and efficiency.

Thus, suppose a boiler designer has to design a special boiler having 90 per cent efficiency and the place where the boiler is set up is such that the tubes can not be more than 10 feet long, what size tubes should the designer use? Take the curve of constant length of 10 feet and follow it to the intersection point with the horizontal line of 90 per cent efficiency; from this point drop a vertical line to the scale at the foot of the chart. The diameter is found to be a little over 1 inch. The designer should use 1-inch tubes.

Again, suppose that it is desirable to lengthen by 6 feet the grate and combustion space of a locomotive boiler without increasing the total length of the locomotive. At the same time it is required that the efficiency should be raised about 5 per cent. Under such conditions the necessary furnace length can be gained by shortening the tube part of the boiler by 6 feet and by decreasing the diameter of the tubes. At present the length of the tubes is 14 feet and the diameter is 2 inches; the tube length of the modified boiler will then be 8 feet. What should be the diameter of tubes if the efficiency is to be raised from 80 to about 85 per cent? To solve this problem, take the curve of constant length of 8 feet and follow it to its inter-

section point with the horizontal line of 85 per cent efficiency; this intersection point comes very close on the vertical line of 1 inch diameter. For the modified boiler, then, the tubes should be 8 feet long and 1 inch in diameter.

Although the authors are confident that the relations shown by the curves of figures 64 and 65 are fairly close to the truth, nevertheless they wish to emphasize that the curves are presented only to illustrate how the dimensions of fire-tube boilers could be easily determined with mathematical accuracy for any desired true boiler efficiency, if the exact relations for these sizes of tubes were first determined by experiment. At present there is no boiler designing based on the principles of heat transmission. The various important relations are merely guessed at, and the boiler builder sells his boilers with guaranteed efficiencies which he is never sure can be realized. As a result of this guesswork, one often comes across results of commercial tests that could not be duplicated and would not stand a close examination.

It may be objected that tubes of small diameter will become choked with soot and cinders. This objection was probably first made by the builder of the stationary fire-tube boiler; still the locomotive boiler builder in the attempt to increase the heating surface used tubes of half the diameter commonly found in stationary boilers; the proof that this reduction of tube diameter was "practical" and that there is no serious trouble from the choking of the small tubes is that thousands of locomotives are in use. Furthermore, the flues of well-kept boilers are cleaned in some plants every day, so that there is little chance for them to get choked.

What has been suggested for the fire-tube boiler can be applied equally well to water-tube boilers. Here, too, the principles of heat transmission can be applied to increase the efficiency. To quicken the rate of heat flow from the gas to the dry surface bring the gas close to the metal by reducing the gas spaces between the tubes through the use of smaller tubes placed closer together; in other words, reduce the "hydraulic mean depth" of the gas passages. For the parallel-flow type of boiler the relation between the efficiency and the "hydraulic mean depth" is probably the same as for fire-tube boilers. An exact determination of this relation would certainly be of great value to the steam-engineering industries. In the cross-flow types of boiler the relation would very likely be complicated by an extra factor, the eddying caused by the expansion and contraction of the gases.

In general, it can be said for all boilers that whatever increases the ratio of the length of the gas path to the "hydraulic mean depth" of the cross section of the path, increases the efficiency of the boiler. This ratio can be increased either by increasing the length of the gas path or by reducing the "hydraulic mean depth."

The length of the gas path can be increased either by increasing the length of the boiler, or by baffling in such a way that the gases will pass successively through different parts of the boiler; in other words, by putting parts of the heating surface in series with one another.

In the water-tube type of boiler, the use of tubes of small diameter would tend to increase the work of cleaning the boiler internally. In plants having condensers this objection would not be serious, and in small plants without condensers the scale-forming ingredients could be chemically precipitated and removed from the water before feeding it into the boiler.

APPLICATION OF THE LAWS OF HEAT TRANSMISSION TO STEAM BOILERS ALREADY INSTALLED.

The application of some features of the laws of heat transmission, deduced from the experimental work of the technologic branch of the United States Geological Survey and the investigations of others, to steam boilers already installed is considered here. It is not practicable in the case of an existing plant to take advantage of all the improvements that the principles of heat transmission suggest. For example, it is unpractical to replace a small number of large flues in a multitubular boiler by a large number of small ones, or to exchange the large tubes of water-tube boilers for a larger number of small tubes placed very close together; in other words, it is not feasible to increase the efficiency of such boilers as heat absorbers by decreasing the "hydraulic mean depth" of the gas passage. On the other hand, it is possible to increase the output of a boiler already installed, by pushing through it a larger quantity of gases; or, it is possible to increase its efficiency by inserting baffles in such a way as to put parts of the boiler's heating surface in series. The authors have noted an application of each of the possible methods, just mentioned, of utilizing more fully the heating surfaces of a boiler to obtain higher capacity and higher efficiency. The plans in each case are so original in conception that they deserve more than casual mention and are therefore discussed somewhat in detail.

INCREASING THE CAPACITY OF A BOILER BY FORCING THROUGH A GREATER WEIGHT OF GASES. WORK OF STOTT AND FINLAY.

H. G. Stott and W. S. Finlay, jr., of the Interborough Rapid Transit Co., New York City, have added a stoker under the rear end of each of several Babcock & Wilcox boilers. This additional stoker permitted the burning of nearly twice as much coal, resulting in twice the weight of gases of combustion being pushed through the boiler at nearly twice the velocity, and making the boiler absorb nearly twice as much heat as when served with a single stoker. The overall efficiency decreased very little.

The steam-generating apparatus used in these tests is shown in figure 66, equipped with two stokers. The boiler is a standard Babcock & Wilcox of cross-flow construction; it was originally served only with the front stoker, which is a Roney. The stoker added in the rear is of the same make, but somewhat smaller. Table 14 gives the principal dimensions of the boilers and furnaces on which comparative tests were run.

TABLE 14.—*Principal dimensions of boilers and furnaces used in Stott-Finlay tests.*

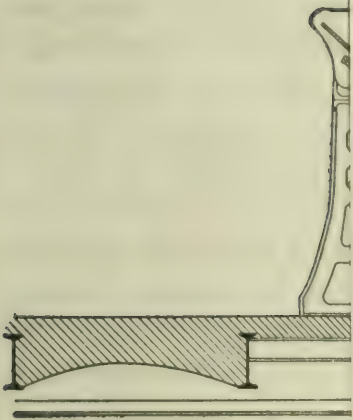
	Single stoker.	Double stoker.
Heating surface.....square feet..	6,008	6,008
Grate area:		
Front.....do....	100	100
Rear.....do....		80
Area of flue at breeching.....do....	20	20
Height of stack above grate:		
Front.....feet..	218	218
Rear.....do....		222
Area of stack:		
Base.....square feet..	201	201
Top.....do....	177	177
Boilers to a stack.....number..	12	12
Approximate combustion space.....cubic feet..	570	1,160

The tests with the double-stoker boiler were made during September and October, 1907. The results are compared with tests made during the first five months of the year 1905 on a similar boiler equipped with a single stoker. Abstracts of the data and results are given in Tables 15 and 16.

TABLE 15.—*Abstracts of data and results of tests on single-stoker boiler.*

Test number.	Total dry coal burned per hour.	Pressure drop (in inches water) from atmosphere to—		Flue-gas temperature.	CO ₂ in flue gases. ^a	Boiler horse-power developed.	Square feet of heating surface per boiler horse-power.	Ordinary boiler efficiency.	B. t. u. presumably liberated per hour.
		Furnace. ^a	Uptake.						
	<i>Pounds.</i>			<i>° F.</i>	<i>Per cent.</i>			<i>Per cent.</i>	
21	1,693		0.104			486	12.37	66.10	24,500,000
60	1,507		.148			478	12.57	73.83	21,560,000
57	1,376		.152	596		429	14.00	73.50	19,420,000
56	1,630		.202	601		516	11.64	74.22	23,200,000
49	1,661		.240	572		511	11.77	72.07	23,650,000
20	1,882		.249			570	10.54	72.21	26,310,000
50	1,843		.306	613		533	11.27	68.95	25,760,000
51	1,688		.348	623		489	12.28	68.85	23,680,000
52	2,064		.399	706		572	10.50	65.83	29,000,000
53	1,955		.441			559	10.75	67.61	27,500,000
61	1,908		.466			549	10.94	65.73	28,840,000
58	2,275		.543			634	9.47	65.40	32,330,000
15	2,409		.602			654	9.18	63.46	34,320,000
16	2,638		.654			793	7.58	68.76	38,400,000
17	2,378		.703			642	9.36	62.36	34,320,000
18	2,600		.752			703	8.55	63.40	37,920,000
59	2,588		.908			674	8.92	60.61	37,080,000
19	2,454		.830			651	9.24	62.10	34,910,000

^a Not recorded.



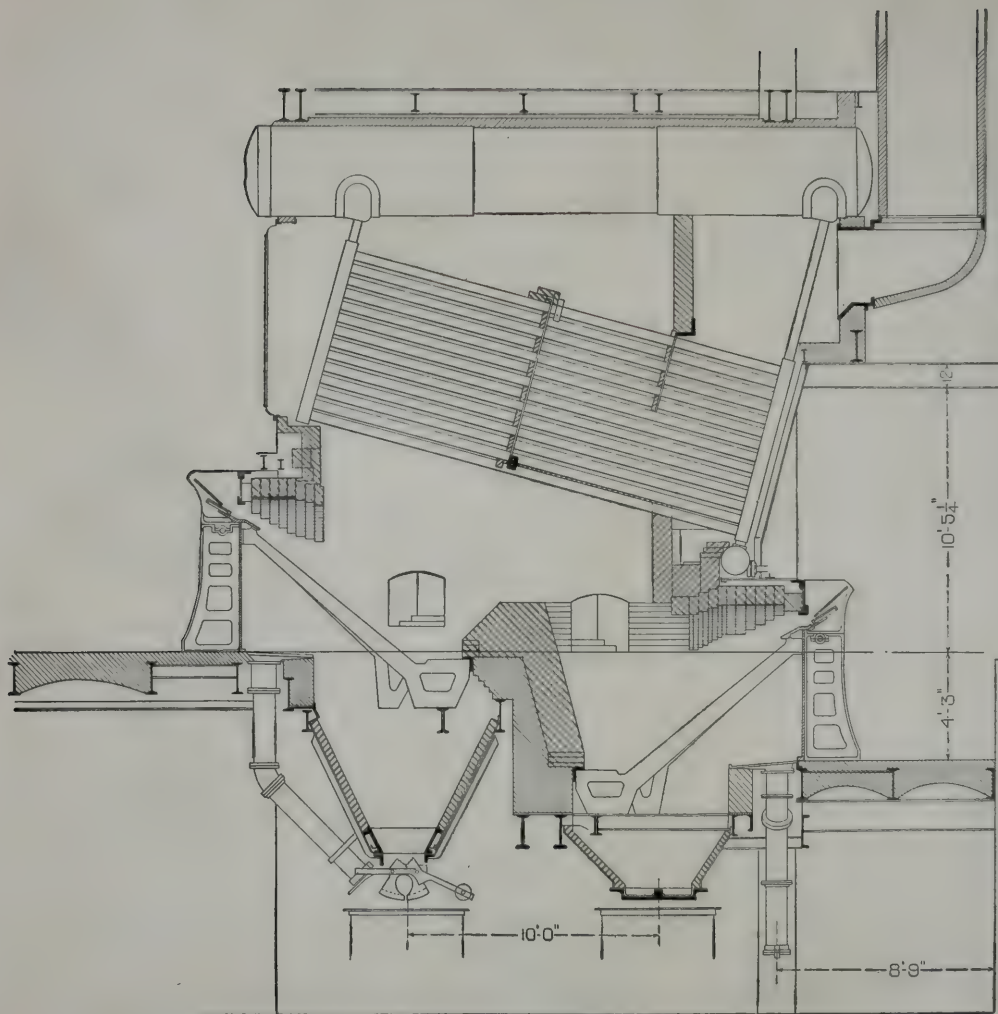


FIGURE 66.—Double-stoker boiler of the Interborough Rapid Transit Co. of New York City.

TABLE 16. *Abstracts of data and results of tests on double stoker boiler.*

Test number.	Total dry coal burned per hour.	Pressure drop (in inches water) from atmosphere to—		Flue gas temperature.	CO ₂ in flue gases.	Boiler horse-power developed.	Square feet of heating surface per boiler horse-power.	Boiler efficiency.	H. T. U. presumably liberated per hour.
		Furnace.	Uptake.						
	<i>Pounds.</i>			<i>° F.</i>	<i>Per cent.</i>			<i>Per cent.</i>	
90	9,598	0.071	0.204	688	11.30	856	7.02	64.04	44,000,000
99	9,478	.084	.251	675	10.63	936	6.92	68.70	45,400,000
102	10,211	.100	.300	730	11.23	937	6.91	64.60	48,300,000
98	10,285	.113	.301	666	12.44	975	6.17	66.07	49,100,000
93	10,675	.112	.352	700	10.83	1,005	5.98	60.04	50,600,000
101	10,732	.123	.400	742	10.40	1,025	5.86	66.65	51,200,000
97	11,387	.127	.402	736	10.70	1,042	5.77	64.48	53,800,000
96	10,592	.156	.404	796	11.30	980	6.13	64.20	50,900,000
94	12,241	.184	.502	814	11.03	1,101	5.46	62.02	59,200,000
100	12,091	.193	.563	763	10.40	1,145	5.22	66.18	57,000,000

DISCUSSION OF RESULTS OF TESTS ON SINGLE AND DOUBLE STOKER BOILERS.

RELATION OF CAPACITY AND EFFICIENCY TO RATE OF HEAT LIBERATION.

The upper curve of figure 67 shows the relation between the boiler horsepower developed and the quantity of heat presumably liberated in the furnace per hour and delivered to the boiler. The quantity is computed from the weight of coal fired per hour and the heat value of 1 pound of coal. The heat liberated is used in preference to the rate of combustion, because the effect of the difference in the quality of coal is thereby eliminated. The hollow points in the figure represent results of tests with the single stoker, and the solid points represent results of tests with the double stoker. The points lie fairly close to the straight line drawn through them, indicating that the output of the boiler increases in direct proportion to the quantity of heat delivered to the boiler.

In the same figure the lower group of points is intended to show how the rate of heat delivery to the boiler affects the efficiency of the steam-generating apparatus. It should be understood that the phrase "boiler efficiency" as it appears on the chart means the combined efficiency of the boiler and the furnace. The points representing the single and those representing the double stoker are so located that it is difficult to draw a single curve through them. As the quantity of heat liberated increases the points representing the single stoker show a strong tendency to drop 4 or 5 per cent below those representing the double stoker. One cause of this drop is undoubtedly the fact that the fuel on the original (front) stoker does not burn completely on account of the small combustion space above that stoker. By adding the rear stoker the combustion space is more than doubled, and much better combustion probably results.

As seen in figure 66, the distance from the front grate to the entrance into the tubes is only about one-third the distance from the rear grate to the same entrance. It is therefore very likely that the gases from the rear stoker are more completely burned. It is on account of this better combustion that, with heavy loads on the boiler, the efficiency is higher when two stokers are used. The point to be emphasized is that any fluctuation in the over-all efficiency is due more to the variation of the efficiency of the furnace than to

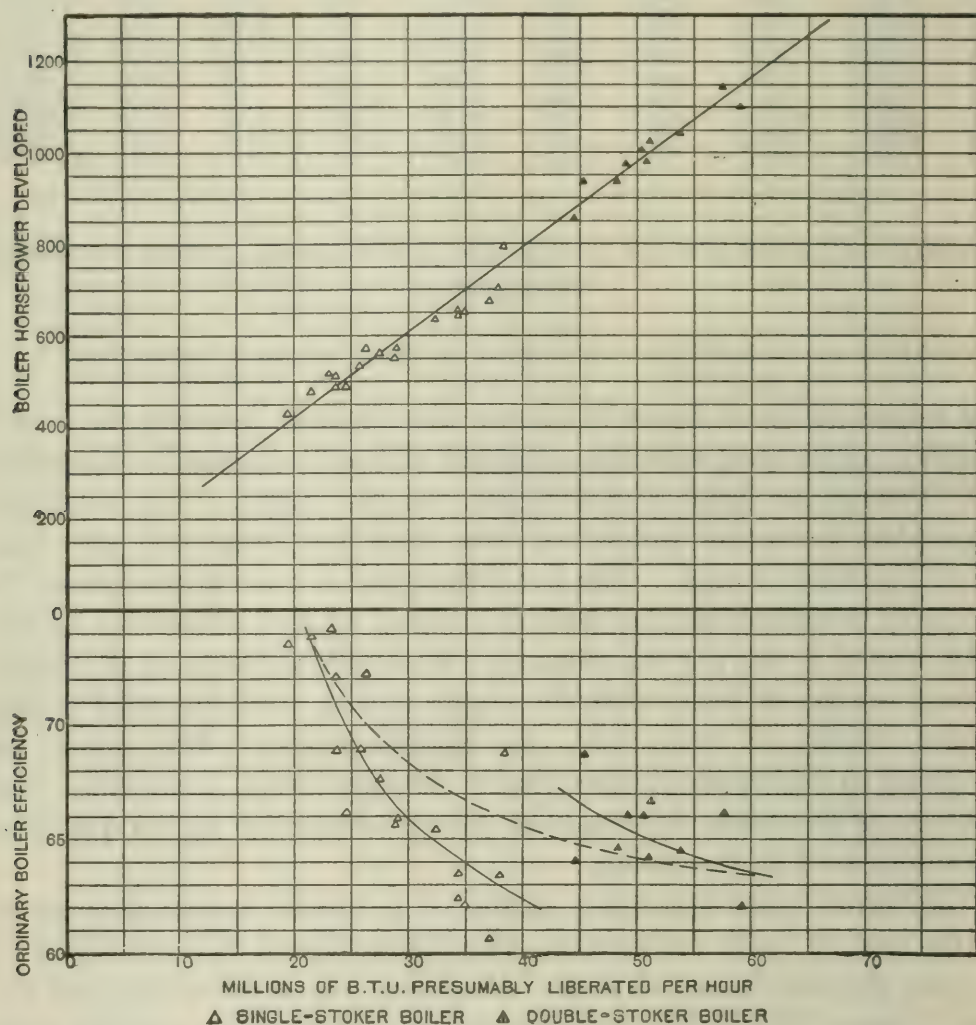


FIGURE 67.—Single and double stoker boilers compared as to capacity and efficiency (ordinary over-all efficiency from coal to steam).

that of the boiler as a heat absorber; the latter efficiency seems to be practically constant and independent of the rate of working.

Another cause of the drop in efficiency shown by the tests with the single stoker below that shown by the tests with the double stoker is that part of the heat liberated on the rear stoker is absorbed by the boiler by radiation and conduction through the tile roof so that the gases are not as hot when they enter the tubes as are the products of combustion from the front grate. Any heat which can be given to the boiler by radiation or conduction is that much clear gain.

To make this statement clear let a specific example be taken. Suppose the boiler under consideration has a true boiler efficiency of 90 per cent. This means that from the time the hot gases enter the tubes until they leave the heating surface of the boiler the latter absorbs by convection 90 per cent of the heat in the gases that is available for absorption. Thus, for instance, if the gases enter the tubes at a temperature $2,000^{\circ}\text{F.}$ above that of the steam the boiler will cool the gases by $2,000 \times .90 = 1,800^{\circ}\text{F.}$, wherefore the gases will leave the boiler at 200°F. above steam temperature. Or, if the gases are at $1,000^{\circ}\text{F.}$ when they reach the tubes, the boiler will cool them down to $1,000 - (1,000 \times .90) = 100^{\circ}\text{F.}$ above the temperature of the steam.

Suppose now that the gaseous products of combustion leave the fuel beds of both stokers at $3,000^{\circ}\text{F.}$ above steam temperature. The gases from the front stoker enter the tubes immediately and give most of their heat to the boiler by convection. If the true efficiency of the boiler is 90 per cent, the gases will leave the heating surface at $3,000 - (3,000 \times .90) = 300^{\circ}\text{F.}$ above the temperature of the steam. The gases from the rear stoker pass along the tile roof and give it part of their heat by radiation and convection. This heat is conducted through the tiles and transmitted to the water in the boiler. The result is that when these gases first come into contact with the boiler tubes they are considerably below $3,000^{\circ}\text{F.}$ Let it be assumed that the temperature of these gases has been reduced to $2,500^{\circ}\text{F.}$ above steam temperature when they enter the tubes. If the boiler absorbs 90 per cent of the heat in them, they will leave the heating surface at $2,500 - (2,500 \times .90) = 250^{\circ}\text{F.}$ above steam temperature, or at a temperature 50°F. lower than the gases from the front stoker; consequently the over-all efficiency with the rear stoker is higher.

A caution must be inserted here: In the case of the boilers under discussion it is evident from a consideration of the data that even at low loads they were working above the rates at which the impartation of heat by natural diffusion of the hot particles of gas among the cold ones was an important factor, so that the rate of heat impartation to the tubes kept up almost fully with the increase in the velocity of the gases forced through the boiler; that is, the scrubbing factor was overpoweringly important. Perhaps the boiler was always working on the level portions of the true-boiler-efficiency curves.

Probably in most water-tube boilers of the cross-flow type the velocity of the gases is greater than the critical velocity and eddying is constant, so that an engineer can usually count on doubling his capacity by doubling his grate surface, at practically no loss in over-all efficiency from coal to steam, provided he has available sufficient total pressure drop ("draft").

However, it is not certain that boilers of the parallel-flow type will always retain their high efficiencies if the weight of gases forced through them be doubled, for some of them seem to be working ordinarily in that low-capacity region where the natural diffusion of the hot particles of gas imparts a considerable portion of the heat, and

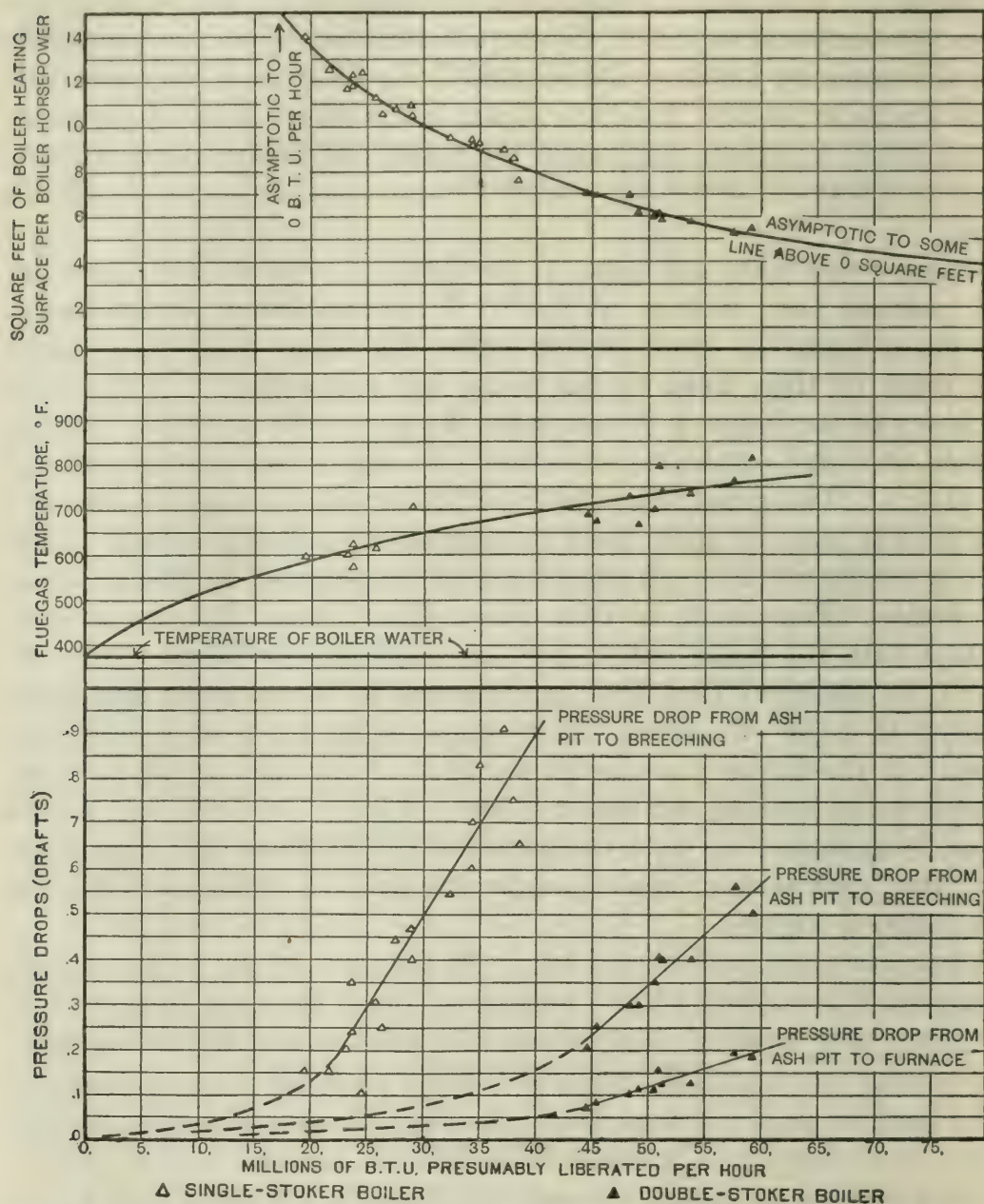


FIGURE 68.—Data concerning Interborough Rapid Transit Co.'s single and double stoker boiler tests.

where the stream lines of flow are not in violent mixture, even though they may not be parallel.

The curve at the top of figure 68 shows that the area of heating surface required per boiler horsepower decreases in the same proportion as the heat delivered to the boiler increases; that is, when the heat delivered is doubled the amount of heating surface required is halved. When 30,000,000 B. t. u. are delivered to the boiler, it takes

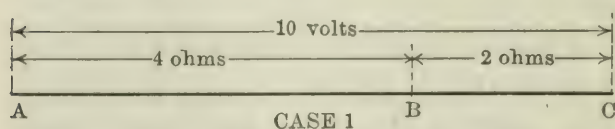
10 square feet of the heating surface to produce a boiler horsepower. When the quantity of heat is doubled, or increased to 60,000,000 B. t. u., the heating surface required per horsepower is reduced to 5 square feet. This relation, of course, naturally follows from that shown by the upper curve of the last figure. The point to be emphasized here is that the area of heating surface required per boiler horsepower can be reduced to any reasonable figure by passing more hot gas through the boiler. At the right the curve is asymptotic, not to the zero line of square feet per boiler horsepower, but to some line above it, because the conductivity of the metal of the tubes is a finite quantity and not an infinite one.

The middle curve of figure 68 indicates that the temperature of the waste gases increases somewhat with the quantity of heat delivered to the boiler. This is perhaps due partly to less complete absorption of heat. It is shown by the lower curve of figure 67 that the "boiler efficiency" is higher for low rates of working; that is, the boiler absorbs slightly more heat from the gases at low than at high rates of working, because the dry surface is at somewhat lower temperatures at the lower capacities.

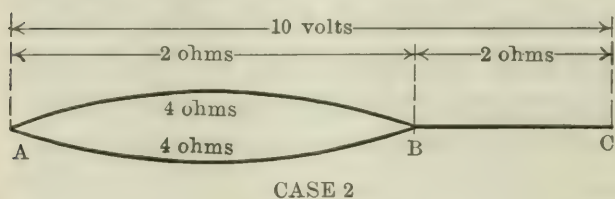
The data for the curves constituting the lower portion of figure 68 are, perhaps, not wholly correct. Since the quantity of heat liberated is proportional to the weight of air passing through the fuel bed, it would seem that the curves passing through the pressure-drop ("draft") points should be parabolas passing through zero and symmetrical with respect to the zero vertical line. The authors have no information as to how these pressure drops were measured nor as to what extent the readings are reliable. Hence it is questionable whether the pressure drops of the single-stoker tests can be fairly compared with those of the double-stoker tests. Inasmuch as the points in each group are consistent among themselves a few remarks are ventured. As the curves are, they assert that in the case of the single stoker a greater total pressure drop is required to push through a smaller weight of gas than in the case of the double stoker. This may be true to some extent, provided the pressure drop through the fuel bed alone is a considerable part of the total pressure drop. It is to be regretted that in the other tests, with the single stoker, the pressures over the fuel bed were not obtained. Such data would have given these speculations much firmer footing.

Suppose, at the rate of heat liberation of 25,000,000 B. t. u., the fuel bed be carried fairly thick so that the resistance to the air passing through it is high, consequently making the pressure drop high. Assume that the pressure drop in that case is 0.2 inch of water or two-thirds of the total drop. Suppose the demand for steam be increased so that the fireman has to liberate 35,000,000 B. t. u. per hour. In the attempt to increase the rate of heat liberation the fire-

man feeds in coal faster and unconsciously increases the fuel bed thickness by 50 per cent, thereby increasing the resistance to gas flow by 50 per cent. He then has to push 140 per cent more gas against 150 per cent of the former resistance. To pass the same weight of gas through twice the thickness of fuel bed requires about twice the pressure drop;^a also, the pressure drop increases in proportion to the square of the weight of gases passing through. There is, then, a pressure drop through the fuel bed in the first case of 0.2 inch of water and a pressure drop in the second case, with the increased output, of



$$0.2 \times 1.5 \text{ (thickness of fuel bed)} \times 1.4^2 \text{ (weight of gas)} = 0.58 \text{ inch of water.}$$



This pressure drop through the fuel bed could easily bring the total pressure drop to 0.7 inch of water as indicated by the curve.

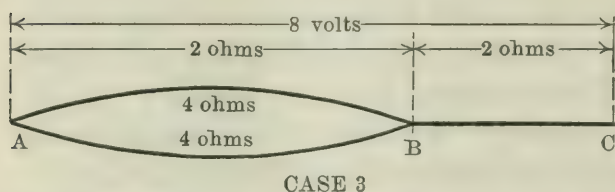


FIGURE 69.—Analogy of the flow of electricity through certain resistances to the flow of gas through fuel beds and boilers.

By adding the rear stoker, the rate of feeding the front one would not have to be changed, so that there would be no likelihood of thickening the fuel bed, and still the rate of heat libera-

tion would increase about 80 per cent; or the rate would rise from 25,000,000 B. t. u. to 45,000,000 B. t. u. per hour. All of this increase would be attained with the same pressure drop through the fuel beds—0.2 inch of water. Of course, the pressure drop through the boiler proper would have to be increased proportionately, but the total pressure drop might be somewhat less in the two-stoker outfit liberating 45,000,000 B. t. u. than in the single outfit freeing 25,000,000 B. t. u. per hour. Addition of grate area can be always looked upon as reduction of resistance to the flow of gas, and will usually result in a proportionate increase of capacity with any given constant pressure drop from ash pit to furnace. The relation between the pressure drop and the output of the steam-generating apparatus with single and double stokers can be illustrated by a similar electrical problem.

Referring to figure 69, let conductor CB (resistance 2 ohms), represent the boiler proper, and the conductor BA (resistance 4 ohms),

^a Bull. 21, Bureau of Mines, p. 30.

represent the fuel bed. If the total drop of potential is 10 volts the current flowing through the two conductors is:

$$\frac{10}{2+4} = 1.66 \text{ amperes.}$$

Suppose another conductor be added parallel to BA and having the same resistance as the latter. The resistance from B to A is then reduced to 2 ohms, or to one-half of what it was before. In this case the current flowing from A to C is:

$$\frac{10}{2+2} = 2.5 \text{ amperes.}$$

Suppose that in a third case the potential drop from A to C is 8 volts. This will make the current flowing from A to C equal to:

$$\frac{8}{2+2} = 2 \text{ amperes.}$$

Although in the last case the total potential drop is smaller than in the first case, more current flows through the system because of the additional conductor from A to B.

In the above problem the electrical resistance of conductor CB is analogous to the resistance of a boiler, and the potential drop and the current may be likened to pressure drop and rate of flow of gases, respectively. As in the above electrical problem, the electrical current flowing through the system can be increased by adding a second conductor to AB; in similar manner the flow of gas through a furnace and a boiler can be increased by adding a second grate. The conditions may be such that more gas flows through the two-stoker apparatus with less total drop than flows through the single-stoker outfit with more drop.

In April, 1911, one of the authors visited the plant where this double-stoker equipment is used and saw 16 boilers with two stokers apiece. Some of these double-stoker boilers had been in daily use over two years and had given entire satisfaction. There had been no trouble from blistering of the tubes.

INCREASING THE EFFICIENCY OF A BOILER BY LENGTHENING ITS GAS PASSAGES. WORK OF W. L. ABBOTT AND A. BEMENT.

Early in 1903 W. L. Abbott and A. Bement, of the Commonwealth-Edison Co., of Chicago, made some comparative tests on a Heine boiler, which they baffled in such a way that the gases passed three times among the tubes of the boiler instead of once, as they do in the case of a boiler with the standard Heine baffling (fig. 70). Figures 71 and 72 illustrate a Heine boiler after the improvement. The triple

gas passage was obtained by inserting two horizontal baffles; the first was laid above the eighth row of tubes and the second above the fourteenth; the total height of the boiler was 17 horizontal rows.

The results of the tests showed that with the addition of these baffles the ordinary boiler efficiency increased from about 57 to

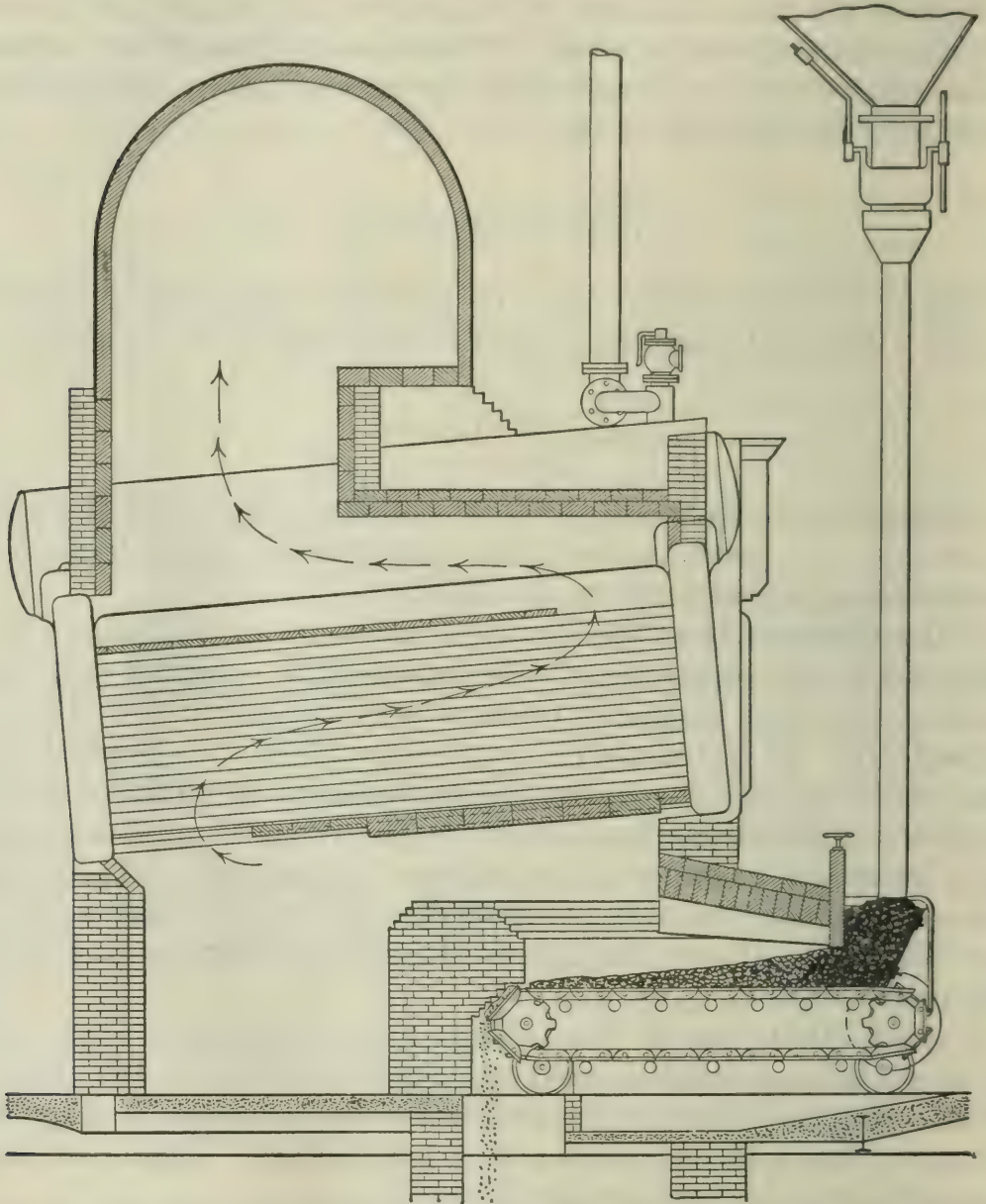


FIGURE 70.—Heine boiler as first installed in the plant of the Commonwealth-Edison Co. of Chicago. The arrows indicate the path traveled by the gases.

about 67 per cent. With approximately the same total pressure drop ("draft") from atmosphere to breeching, the capacity of the rebaffled boiler was somewhat higher than that of the standard one. Table 18 gives an abstract of the data and results of these tests.

Table 17 gives the principal dimensions of the boiler used in these tests.

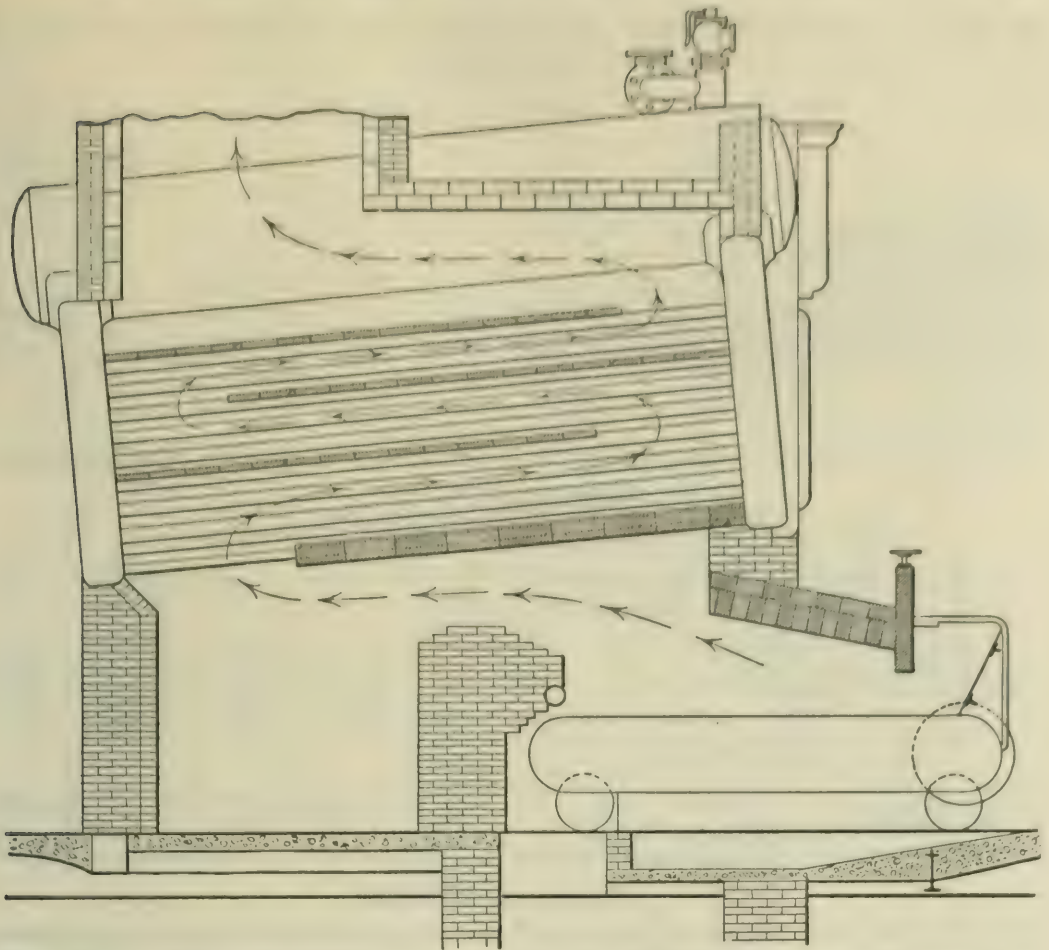


FIGURE 71.—Triple-pass Heine boiler as baffled by W. L. Abbott and A. Bement. Side view. The arrows indicate the path traveled by the gases.

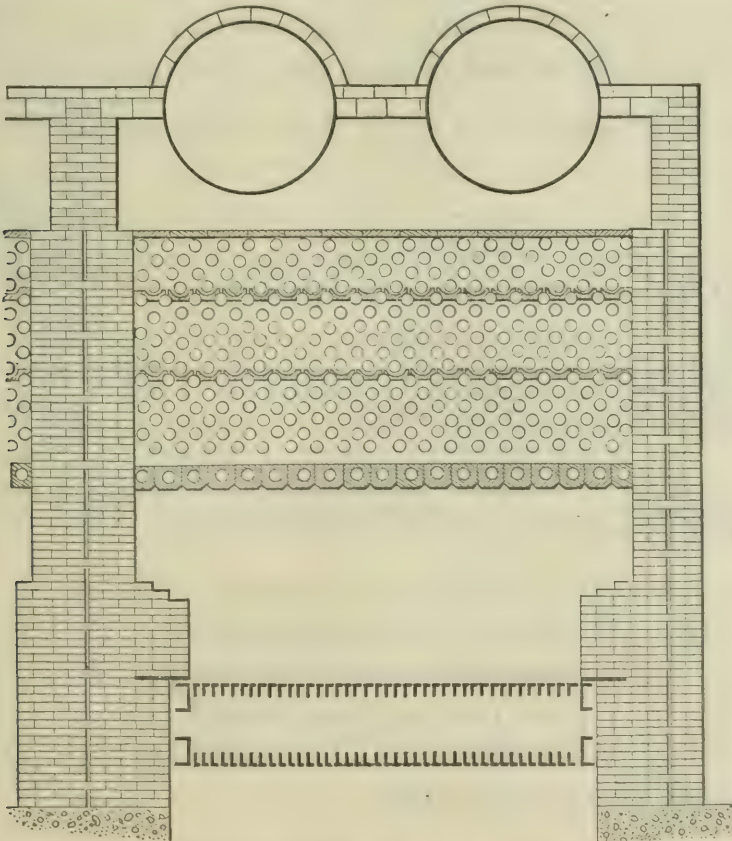


FIGURE 72.—Triple-pass Heine boiler as baffled by W. L. Abbott and A. Bement. End view

TABLE 17.—*Principal dimensions of boiler used in tests of a single and of a triple pass Heine boiler.*

	Single pass.	Triple pass.
Number of tubes high.....	17	17
Number of tubes wide.....	19	19
Outside diameter of tubes.....inches..	3.5	3.5
Length of tubes.....feet..	15.8	15.8
Total water-heating surface, tubes, and drums.....square feet..	4,800	4,800
Normal boiler horsepower, standard rating.....	480	480
Approximate length of travel of gases among tubes.....feet..	13.3	36
Area of B. & W. chain-grate stoker.....square feet..	72.2	72.2

TABLE 18.—*Abstracts of data and results of boiler tests made on a standard and on a triple-pass Heine boiler.*

SINGLE-PASS (STANDARD BAFFLING) BOILERS.

Test No.	Dry coal burned per hour (pounds).	Pressure drop ("draft") from atmosphere to furnace (inches of water).	Temperature (°F.) of—			Per cent of CO ₂ in flue gases.	Boiler horsepower developed.	Total equivalent evaporation per hour.	Ordinary "boiler efficiency."
			Steam.	Furnace.	Flue gas.				
1	1,757	0.38	377		594	7.0	382.9	13,219	54.64

TRIPLE-PASS BOILERS.

1	2,051	0.22	377		442	6.7	395.0	13,629	68.14
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In the fall of 1906 the same experimenters made a series of tests with a similar boiler baffled as shown in figures 73 and 74.

TABLE 19.—*Principal dimensions of a single and of a double pass Heine boiler used in tests.*

	Single pass.	Double pass.
Number of tubes high.....	17	17
Number of tubes wide.....	19	19
Outside diameter of tubes.....inches..	3.5	3.5
Length of tubes.....feet..	15.8	15.8
Total water-heating surface, tubes, and drums ^asquare feet..	4,800	4,800
Normal boiler horsepower, standard rating.....	480	480
Approximate length of travel of gases among tubes.....feet..	13.3	24.5
Area of B. & W. chain-grate stoker.....square feet..	72.2	72.2

^a For the double-pass boiler all the drum surface is included as fully as for the single and the triple pass boiler, although it was scrubbed by only a small part of the gases, this part being admitted through a 2-inch opening round the front end of the top baffle, next the front water leg.

A single horizontal baffle was inserted between the tenth and eleventh horizontal rows of tubes, causing the gases to pass twice through the tube space before entering the breeching, thus coming into contact only with the rear portion of the steam drums; the remainder of about four-fifths of the heating plate of the steam drums was left nearly ineffective.

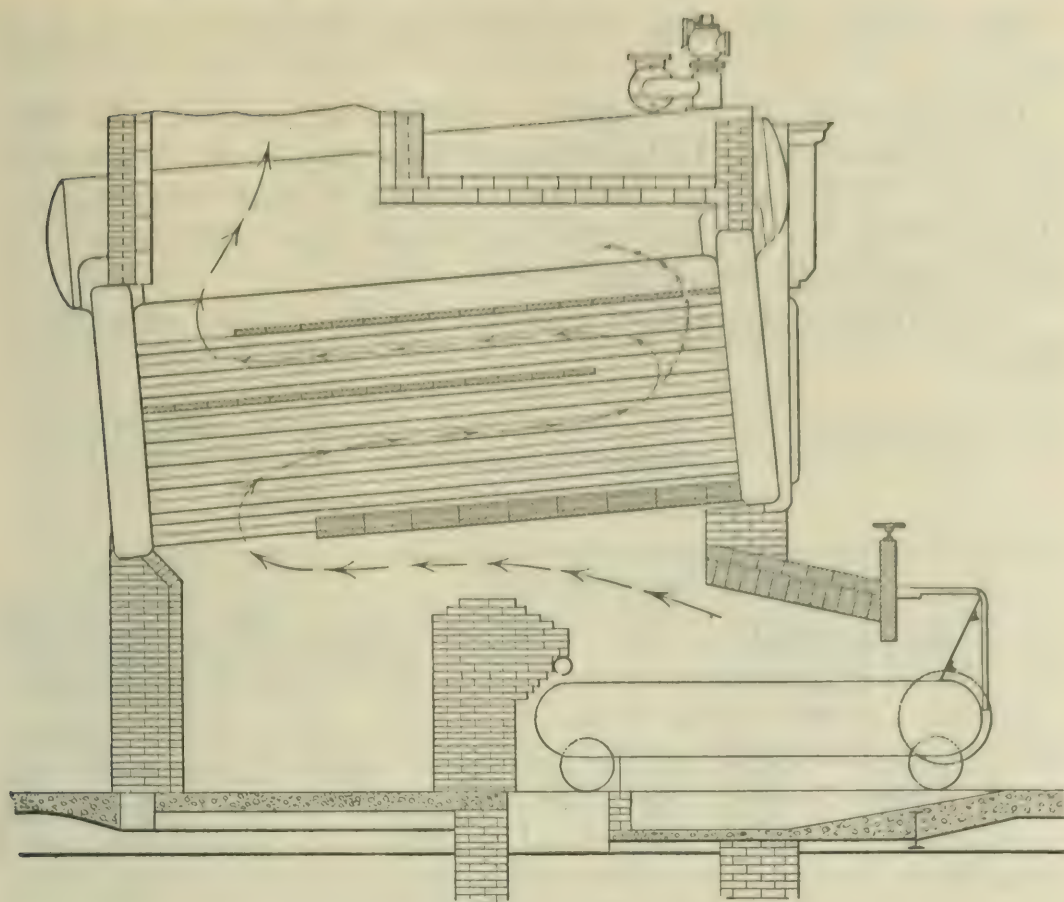


FIGURE 73.—Double-pass Heine boiler baffled by W. L. Abbott and A. Bement. Side view. The arrows indicate the path traveled by the gases.

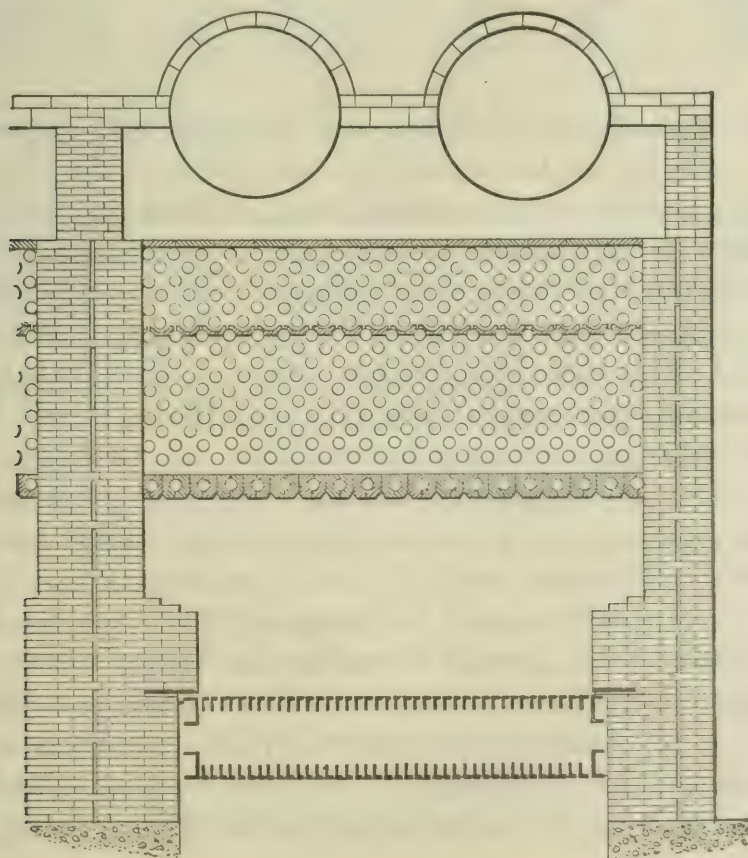


FIGURE 74.—Standard short-length Heine boiler baffled by W. L. Abbott and A. Bement. End view.

The principal dimensions of the boiler used in these tests are given in Table 19.

The results of the tests showed that the addition of this baffle increased the ordinary boiler efficiency from 57 per cent to about 62 per cent. With the same total pressure drop ("draft") the capacity obtained with the rebaffled boiler was about 5 per cent higher than that obtained with the standard boiler.

Table 20 gives abstracts of data and results of these comparative tests.

TABLE 20.—Abstracts of data and results of boiler tests made on a standard and on a double-pass Heine boiler.

SINGLE-PASS (STANDARD) BOILER.

Test No.	Dry coal burned per hour (pounds).	Pressure drop ("draft") from atmosphere to furnace (inch water).	Temperature (°F.) of—			Per cent of CO ₂ in flue gases.	Boiler horse-power developed.	Total equivalent evaporation per hour.	Ordinary "boiler efficiency."
			Steam.	Furnace.	Flue gas.				
2	2,775.7	0.40	379	2,582	621.7	7.25	494.6	17,065.4	52.88
3	2,984.5	.40	379	2,648	651.1	7.63	550.9	19,007.3	53.78
6	2,920.5	.42	379		634.4		567.6	19,585.1	57.68
7	2,816.8	.40	379		653.0		524.0	18,078.4	56.05

DOUBLE-PASS BOILER.

2	2,572.5	0.37	379	2,550	550.0	9.62	512.6	17,685.7	57.43
3	2,739.9	.40	379	2,616	537.8	10.22	568.3	19,606.5	59.18
6	2,715.1	.41	379		603.0		567.6	19,585.1	61.63
7	2,716.7	.39	379		611.5		575.5	19,855.8	62.91

The fact that Messrs. Abbott and Bement left, adjacent to the front water leg, a 2-inch opening the full width of the boiler to allow part of the gases to go around the drum, affords an opportunity for some remarks on "hydraulic mean depth" and the equivalent of the hydraulic engineer's "wetted perimeter." These experimenters are not to be blamed for having left this opening, inasmuch as the fundamental principles of heat absorption were then little understood.

From previous demonstrations in this bulletin, it will be realized that if the upper pass (among the tubes) in the two-pass boiler had not been relieved of part of its gas, it would nevertheless have absorbed almost as large a percentage of the available heat as it did.

The "hydraulic mean depths," or the average mean distances of the particles of gas from the inclosing surfaces, are far less among the tubes than in the passage alongside the drum; moreover, all the surface rubbed by the gas among the tubes is heating surface; on the other hand, most of the surface rubbed by the gas alongside the drum is brick wall and soot-covered floor, and not heating surface. Therefore, much less heat was absorbed from the gases passing under the

drum than would have been absorbed if they had passed with the bulk of the gases through the upper pass among the tubes.

By inserting additional baffles in the tube space and reducing the cross section of the gas passage, the "hydraulic mean depth" was not changed, and the average distances of the gas particles from the heating surface remained the same. For the same reason, if, in the cross-flow type of boiler, the three-pass flow is changed by the addition of vertical baffles into a five pass, the "hydraulic mean depth" is not changed. Any increase in the efficiency in that case is solely due to the increased length of the gas path.

DISCUSSION OF RESULTS OF TESTS ON A HEINE BOILER WITH SINGLE, DOUBLE, AND TRIPLE GAS PASSAGES.

The addition of one or two horizontal baffles rearranges the heating surface of the tubes. In the standard boiler the whole heating surface of the tubes is arranged in parallel with respect to the gas travel. When one horizontal baffle is inserted, about two-fifths of the total heating surface of the tubes is placed in series with the remaining three-fifths and the length of the gas passage is nearly doubled. By the insertion of two horizontal baffles the heating surface of the tubes is separated into three unequal parts which are arranged in series with one another in the path of the gas stream. This latter modification nearly triples the length of the gas path of the standard baffling. In the double and triple pass boilers the section of the tubes forming the chamber that the gases enter first is made considerably larger than the succeeding section or sections because the gases when entering this first section are much hotter and therefore occupy much larger volumes than when passing through the following sections.

With nearly the same total pressure drop ("draft") the velocity of gas through the rebaffled boilers is somewhat higher than through the standard boiler. This conclusion is deduced from the fact that with all three types of baffling the rate of combustion of the coal and the quantity of air used were nearly the same, resulting in nearly the same rate of gas generation. Since the cross sections of the gas passages in the rebaffled boiler are smaller, the gases must flow faster than they do through the larger passages of the standard boiler. If accurate measurements of the gas pressure among the boiler tubes had been taken, undoubtedly it would have been found that the pressure drops through the double and triple pass boiler were appreciably higher than through the standard boiler, otherwise the velocities could not have been greater. Unfortunately only the pressure drops through the fuel beds were recorded during the tests.

The higher efficiency obtained with the rebaffled boilers is due to the increased length of the gas passages. The longer the path of

the hot gas the more contacts each particle of the gas makes with the surfaces of the tubes, and the lower the temperature of the gas when it finally leaves the boiler. A lower temperature of the leaving gases means that the heating surfaces absorb more heat, so that the efficiency of the boiler as a heat absorber is increased.

The increase in boiler efficiency due to the addition of baffles is similar to the increase of efficiency of boiler No. 3 over that of boiler No. 1 in the series of laboratory boiler tests discussed in a previous portion (pp. 26 to 80) of this bulletin. Boiler No. 3 can be considered as boiler No. 1 with gas passages made twice as long. In fact, boiler No. 1 could be made to have the same efficiency as No. 3 by erecting baffles in such a way that the hot gas would pass first through six of the flues in one direction and then return through the remaining four.

The approximate averages of the initial and final temperatures of the Abbott-Bement tests on the single, double, and triple pass boilers are platted as ordinates on the lengths of the gas passage as abscissæ and are given in the lower curve of figure 75. The initial temperature was measured on only some of the tests; however, judging from the CO_2 contents in the flue gases the assumed initial temperatures are perhaps very close to the actual temperatures of each test. The flue-gas temperatures as measured by the flue thermometers are all too low because no attempt was made to screen the thermometer bulbs from radiating heat to the cold surfaces of the boiler and breeching. Still, as the flue thermometers in all the tests were inserted at similar places in the boiler setting, the temperatures indicated are comparative.

The temperature curve of figure 75 is similar to the curves of figure 16, the similarity being particularly shown in the lowest curve. Comparison of the curves of the two figures at once suggests the likelihood that the flue temperatures as given in figure 75 are too low and that the apparent drop of temperature through the one-pass boiler or through the first pass of the other boiler is too steep.

The curve of flue temperature beyond the point for the single-pass boiler approaches a straight line and becomes nearly horizontal. On the assumption that the flue temperatures are at least relatively correct, the curve indicates that it would hardly pay to lengthen the path of the gases more than by the double or triple pass; any further increase of the gas passage would reduce the flue-gas temperature very little. The upper curves of figure 75 show the increase in ordinary "boiler efficiency" when the boiler is baffled to form a double or triple gas passage. Two curves have been drawn because on the two sets of tests with the single and double pass boilers the efficiencies obtained differed somewhat, as shown in Table 18. This difference in efficiency in the two sets of tests was probably due to varying quality of

coal. The efficiency for the triple-pass boiler is approximated from the flue-gas temperatures.

As the ordinate of the curves is the ordinarily used "boiler efficiency," and not the true boiler efficiency, the curves are not asymptotic to the 100 per cent line, but to a horizontal line somewhat lower,

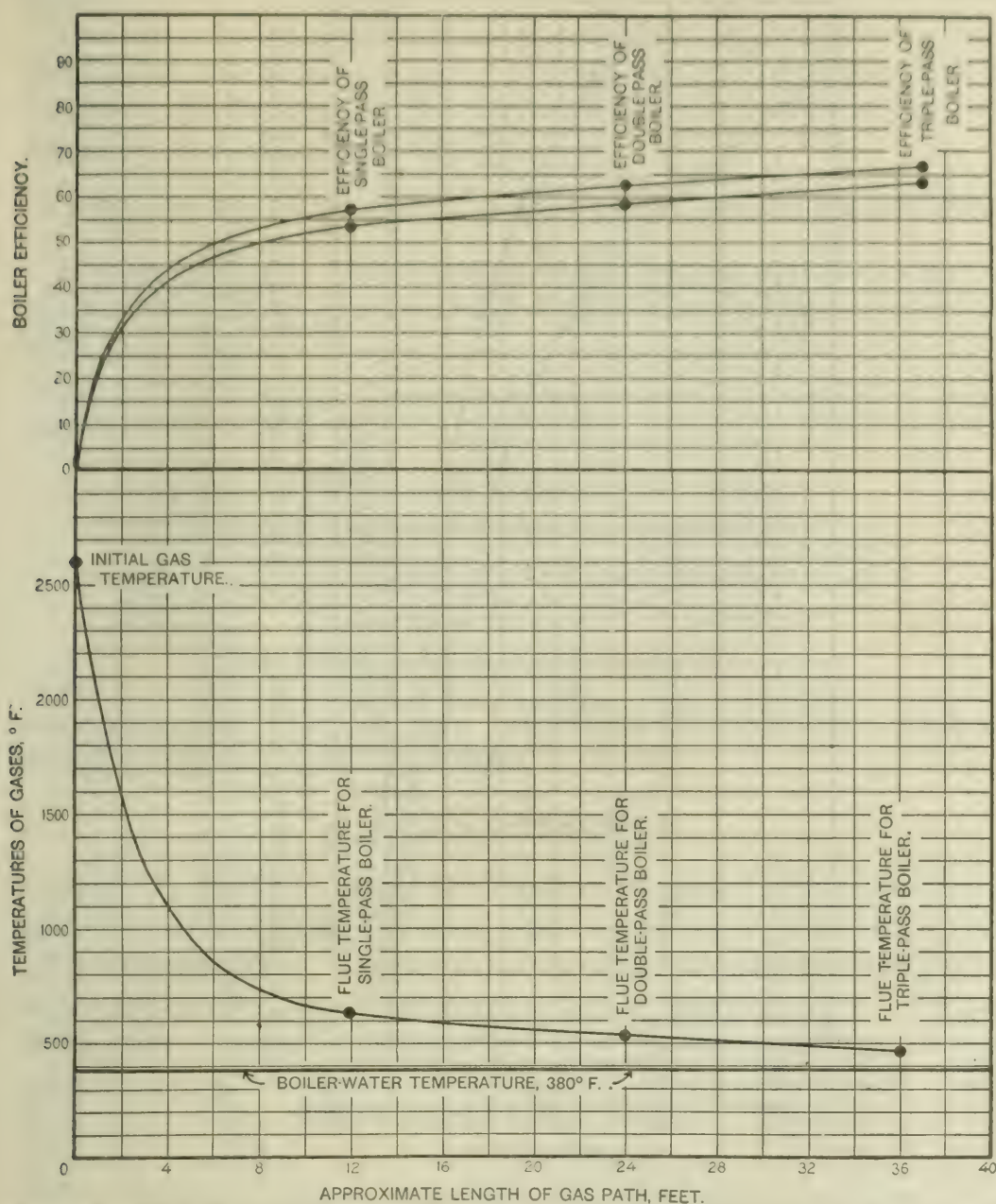


FIGURE 75.—Effect of change of length of gas travel on efficiency and flue temperature in a parallel-flow water-tube Heine boiler. Abbott-Bement tests.

perhaps at about 80 or 85 per cent, for the reason that the heat in gases below steam temperature can never be absorbed by the boiler.

The gain in efficiency of the triple-pass boiler over the double-pass is not as large as the gain of the double-pass over the single-pass, and the gain would grow smaller for each succeeding increase in length of gas passage through baffling. One must admit this fact if he considers that it would take an infinitely long passage to obtain the

limiting efficiency of 80 or 85 per cent. Therefore in increasing the length of the gas passage to obtain higher efficiencies, a limit is reached beyond which it would not pay to work the boiler. There are perhaps few boilers in use at present which work so near this limit that they could not be improved by lengthening the gas travel.

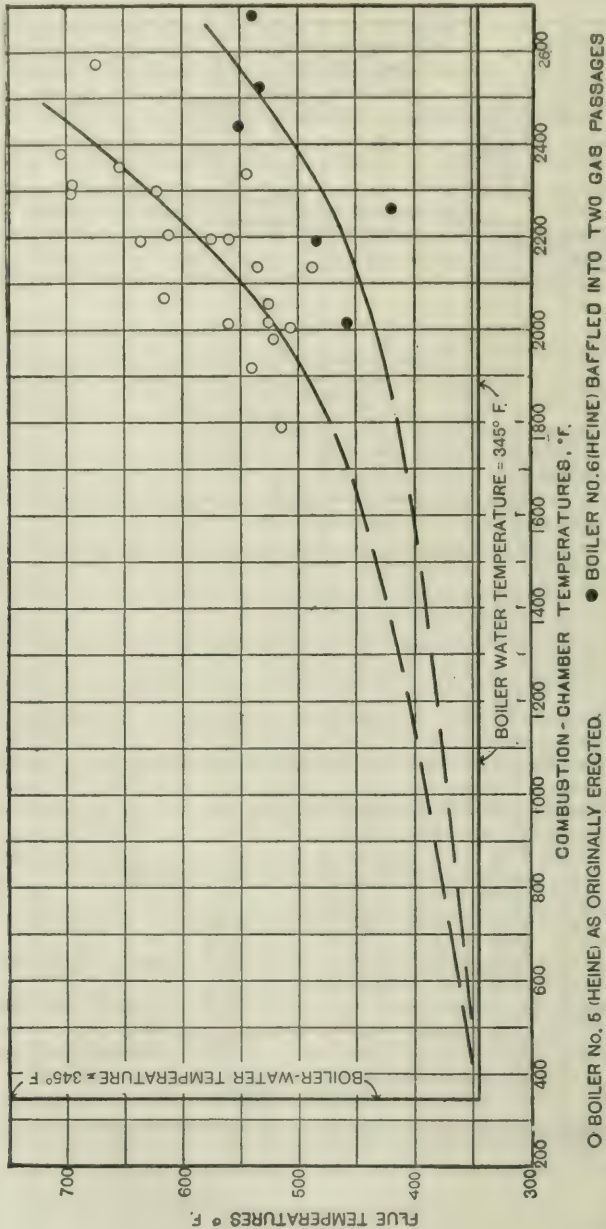


FIGURE 76.—Relation of flue-gas and combustion-chamber temperatures in a single and a double pass Heine boiler in the former plant of the United States Geological Survey at Norfolk, Va.

WORK OF THE UNITED STATES GEOLOGICAL SURVEY.

In the Norfolk, Va., plant of the Geological Survey one of the two 210-horsepower Heine boilers had been rebaffled according to the Abbott-Bement method, so as to cause the gases to pass through the tubes twice instead of once. The result was that about 8 per cent more heat was absorbed, with no reduction in capacity when the same total pressure drop ("draft") was used. When the pressure drop was raised and the boiler was made to produce about twice its rated amount of steam it still retained nearly all of the increased

efficiency. Here is an instance of greatly increasing the output at a slight cost and at the same time increasing the evaporation per pound of coal.

Figure 76 shows the relation of the combustion-chamber and the flue-gas temperatures for the tests made at the Jamestown Exposition plant.

The upper curve gives this relation for the standard or single-pass boiler, and the lower curve for the rebaffled or double-pass boiler. As shown by the curves, the double-pass boiler cools the gases considerably more than the single-pass one.

APPLICATION OF SURFACE COMBUSTION TO STEAM BOILERS.

If a stream of a mixture of combustible gas and air is directed against the surface of a heated body, the mixture burns rapidly with little or no flame. The combustion takes place at the surface of the body, or within a short distance of it, and is therefore termed "surface combustion." Although the combustion is quicker if the body is of

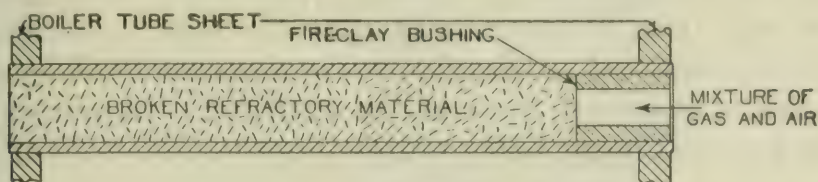


FIGURE 77.—Cross section through tube of experimental boiler.

some refractory material, any noncombustible substance, such as a metal, produces this phenomenon.

BONE'S EXPERIMENTS.

Prof. W. A. Bone, of the University of Leeds, England, within the last two or three years has done considerable experimental work with surface combustion. One of the most interesting features of his work is his application of surface combustion to steam boilers. Professor Bone used a specially constructed fire-tube boiler having 10 tubes, each 3 inches in internal diameter and about 3 feet long. In the front ends of the tubes were fire-clay bushings $1\frac{1}{2}$ inches in internal diameter and about 4 inches long. The remainder of the length of the tubes was completely filled with finely broken refractory materials.

Figure 77 shows a cross section of one of the tubes. In starting this boiler, gas and air are brought into the fire-clay bushing separately to avoid back-firing. After the broken refractory material has been brought to a red heat, the gas and air are admitted as a mixture with little excess of air, and combustion goes on indefinitely. The broken refractory material in the front part of the tubes becomes incandescent, but toward the rear it remains dark and comparatively cool.

With this apparatus, Bone obtained an evaporation of over 20 pounds of water per average square foot of the tube surface per hour. However, the front foot of the length of the tubes caused about 60 per cent of the total evaporation, whereas the last foot of the length of the tubes caused only about 10 per cent.^a

The significant feature of this experiment is that the convection of heat is largely, although not entirely, replaced by radiation and conduction. Instead of the heat being developed in the furnace and carried by the gases several feet to the heating plates of the boiler, it is developed on the surfaces of the broken refractory material within a fraction of an inch from the metal surface of the tubes. The small portion of the heat contained in the gaseous products of combustion is taken up rapidly by the refractory material as the gases pass through in very fine streams. The individual particles of the refractory material are in contact with each other or with the metal of the tubes so that the heat travels mostly by conduction from the central portions of the tubes to the metal surface. Any small interstices are readily crossed by radiation. The rate of convection of heat from the moving gaseous products to the surfaces of the refractory material is brought to the topmost limit by bringing the particles of gas very close to the surfaces. On account of the refractory material being finely crushed and the particles lying close together, the interstices are very small, so that the average distance of the gas particles or molecules is a very small fraction of an inch. Added to this is the irregularity of the interstices, which causes considerable eddying in the small streams of gases. Consequently the gaseous particles make contact with the surfaces rapidly, causing a rapid rate of heat impartation. The interstices between the particles of refractory material are equivalent to tubes of very small internal diameter and have the advantage that the very small passages can be obtained cheaply as compared with making boilers with very small tubes.

It has been pointed out several times in this bulletin that by far the slowest of the processes involved in the transfer of heat from the burning fuel to the boiler water is that from the gaseous products of combustion to the dry surface of the boiler heating plates. The occasion for this process Professor Bone almost entirely avoids. His filling the boiler tubes with solid heat-conducting and heat-radiating material is practically equivalent to extending the dry surface of the heating plates, as defined in this bulletin, to the center of the tubes. The path of heat travel is shown diagrammatically in figure 78. In the diagram A and B represent the average point of location of the heat, which in reality is a cylinder of average distances of all the

^a For a somewhat more detailed account of these tests see "Power," November 21, 1911, p. 767.

points in the circles of all cross sections through a tube. Figure 78 is somewhat similar to figures 1 and 46. In figure 78 the radiation is shown by a discontinuous line to indicate that the radiation is in most part from one particle of refractory material to another. It is difficult to construct a diagram that will exactly represent the real condition; however, figure 78 may help in studying, or suggest a

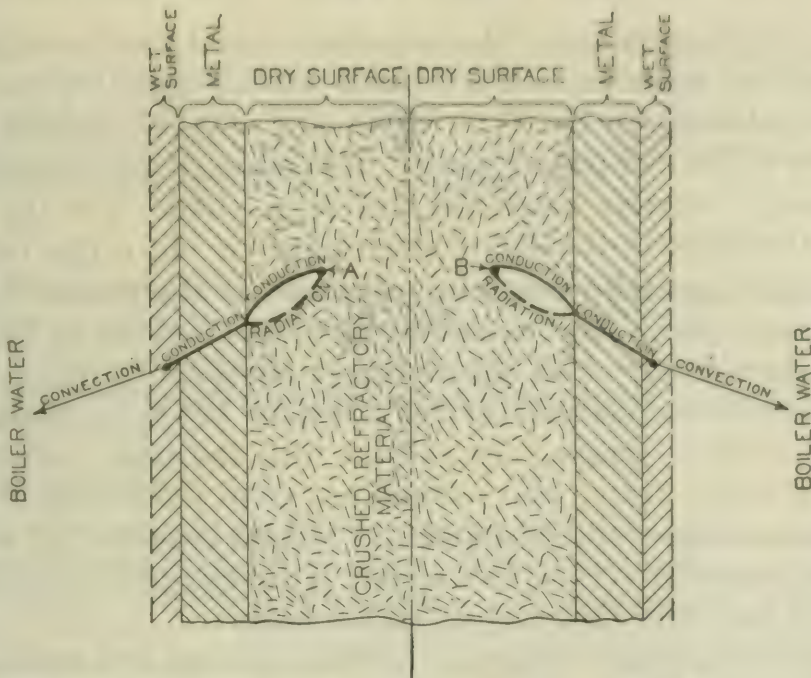


FIGURE 78.—Cross section of tube of experimental boiler, showing path of heat travel.

method of studying, the process of heat transmission in the boiler tubes.

The results of Bone's experiment demonstrate the statement made several times previously that the heating plates of a boiler will transmit all the heat that can be delivered to them.

It is difficult at this time to estimate the effect of Bone's results on the future design of apparatus. Surface combustion can not be applied to solid nor perhaps to liquid fuels. It is not impossible that in future gas producers of high capacity will be designed that will efficiently gasify solid and liquid fuels, which can then be used in steam-generating apparatus designed to take advantage of the benefits derived from the principles of surface combustion.

APPENDIX.

APPLICATION OF THE VELOCITY FACTOR TO PROCESSES OTHER THAN STEAM GENERATION.

During the prosecution of the experiments and the development of the theoretical treatment of the theme of this bulletin, the writers called the attention of Dr. C. S. Hudson, a physical chemist then in the employ of the fuel-testing plant. He became much interested and offered many valuable criticisms and suggestions. For the mathematical calculation at the end of this chapter credit is due to him.

The feature specially deserving comment is the probable general bearing of the "scrubbing" or mixing factor pointed out by Prof. John Perry on the velocity of all heterogeneous chemical reactions. Many investigators have done work in this direction, but the authors believe that none of them grasped the subject with the same mathematical firmness as Prof. Perry, who worked out from the kinetic theory of gases the postulation that the quantity of heat imparted by a hot gas to a (clean) metallic surface in a second is directly proportional to the product of the three first factors stated below:

- (1) The temperature difference between the gas and metal.
- (2) The density of the gas, reckoned in terms of the number of gram-molecules per unit volume of the gas.
- (3) The velocity parallel to the metallic surface of that portion of the gas adjacent to the metallic surface, the lines of flow next the surface being assumed to be straight.

This approximately linear relation between the velocity and the quantity of heat transmitted, as first suggested by Prof. Osborne Reynolds and later developed mathematically by Prof. John Perry, is so remarkably simple as to suggest the query whether or not the same straight-line law is in general applicable to all heterogeneous physical and chemical reactions. The following instances appear to indicate an affirmative answer to the query.

One of the most striking instances that are of particular interest to engineers is the straight-line relation between the rate of combustion of fuel on the grate and the velocity of air passing through the fuel bed. Everybody is familiar with the fact that as the "draft" is increased the rate of combustion increases. This increase in the rate of combustion is directly proportional to the velocity of air through the burning fuel. As a practical illustration of this fact the writers refer to Bureau of Mines Bulletin 21, page 52. The upper curve of the chart on that page shows that the rate of combustion is a straight-line

function of the weight of air supplied per hour. The velocity through the fuel bed is directly proportional to the weight of air supplied, so that this straight-line relation exists between the velocity and the rate of combustion.

In Cleveland Abbe's book on "The relation between climates and crops" (U.S. Weather Bureau Bulletin No. 36), this passage occurs on page 105:

The evaporation of water from the leaves and from the ground depends upon the temperature, wind, and humidity of the air. It is a rather complex result; if the above-mentioned elements remain constant for any length of time at the surface of the mass of water the evaporation from that surface will be closely represented by the following formula, which is due to Fitzgerald of Boston,

$$E=0.0166 (P-p) (1+\frac{1}{4}W),$$

where W is the velocity of the wind in miles per hour, P the pressure of the vapor in inches of mercury corresponding to the temperature of the water, p the pressure of vapor corresponding to the dew point in the free air, and E the evaporation expressed in inches of depth of water evaporated per hour under atmospheric pressure between 29 and 31 inches of barometer.

Also on page 106:

Prof. Thomas Russel, of the Signal Office, has published results of some observations on the effect of the wind on the evaporation from the disks of the Piche evaporimeter. (See Annual Report Chief Signal Officer, 1888, p. 176, or Monthly Weather Review, 1888, p. 245.) He finds that with the temperature of the air 84° F., and a relative humidity of 50 per cent, the evaporation varies with the velocity of the wind at the surface of the moist disk as in the following paragraph:

"At 5 miles per hour the evaporation is 2.2 times that in a calm; at 10 miles, 3.8 times; at 15 miles, 4.9 times; at 20 miles, 5.7 times; at 25 miles, 6.1 times; at 30 miles, 6.3 times."

Up to 20 miles per hour this is a linear function of the velocity of the wind; above 20 miles the evaporation falls off.

The Piche evaporimeter consists of a glass tube closed at the top, and fitted at the bottom with a horizontal disk of bibulous paper of about twice the diameter of the tube; the tube is filled with water, the rate of evaporation of which is read off from graduations on the tube. It is evident that at high wind velocities the temperature of the water will become considerably lower, and thus the above experiments are only approximately indicative.

Another instance is found in the inaugural dissertation of E. S. Merriam, Goettingen, 1906. He gives some measurements of the residual currents through salt solutions. Perhaps the subject of residual currents may not be familiar to all readers, so a brief definition is given here. When an electric current is sent by an imposed electromotive force through a salt solution, which at the beginning is of uniform concentration, there is rapidly produced a counterelectromotive force at the electrodes, due to the electromotive force of the concentration difference at the two electrodes. (All this under the condition that the impressed voltage is very low so that no gases are

formed at the electrodes.) This counterelectromotive force would rise in value and finally stop the current if it were not that the electrolyte in the main body of the solution diffuses into the weaker solutions around the electrodes, or vice versa, resulting in reductions of the concentration differences (and total difference between the electrodes), and hence in a reduction of the counterelectromotive force. After the voltage has been impressed for some time a steady strength of current is reached, which is called the "residual current." The final strength of current is obviously determined by diffusion processes, and its magnitude is therefore increased by any causes which make the diffusion faster, stirring, for example. Merriam rotated one electrode, which was small; the other was stationary and so large that no appreciable change of concentration occurred at it. His measurements give the relation between this residual current and the speed of rotation of the electrode. Although he considers the relation to be logarithmic, if the results are platted with the residual current as ordinate and the velocity of rotation as abscissæ, the points will lie as nearly on a straight line as one could wish. The logarithmic equation is only an empirical one, whereas the linear equation is directly analogous to Perry's well-based linear equation for the transmission of heat into a boiler plate from a stream of hot gas, moving parallel to it.

On the basis of the kinetic theory of gases, Perry reasons that the rapidity with which the cold adhering molecules are replaced by hot ones is a direct function of the translational velocity of the gas as a whole past and parallel to the heating surface. There is no reason known to the present authors why the same reasoning will not apply in heterogeneous chemical reactions.

The assumption of the reality of these adhering films is fairly well substantiated. Dr. O. E. Meyer, in his standard book entitled "The Kinetic Theory of Gases," as much as postulates the existence of such a film next the surface of solids. There is also a great deal of evidence from experiments, some of which is given here.

Briggs^a attempted to calculate the thickness of the layer of water adsorbed (held on the surface, not absorbed) on quartz grains of several sizes and concluded that it was about 2.66×10^{-6} cm., if the layer had a uniform density equal to that of water; the mass of grains was exposed to air at 30° C., 99 per cent saturated with water vapor.

It is also likely that water vapor adheres to glass and certain minerals with great tenacity, as they must in some cases be heated to several hundred degrees temperature before water vapor ceases to come off. In these cases, however, one can not be sure that there is not some sort of chemical combination; but if there is, the velocity of the reaction would nevertheless fall more or less under the influence of the "scrubbing" factor.

^a Jour. Phys. Chem., vol. 9, 1905, p. 617.

Müllfarth ^a found that perfectly dry glass powder absorbed ordinary gases and that the weight of the film was governed approximately by the temperature pressure laws of gases. This is about what should be expected, in the absence of chemical hygroscopic effects, if one postulates that the gases are not held as a liquid film of homogeneous density but that among and close to the outer molecules of the solid the gases may be denser than their liquid phases (although in the gaseous state) and that the density decreases outward according to an anti-compound-interest law, with, however, the stipulation that the lower limit of density is not zero but the density of the atmosphere.

More evidence from experiments will not be given, for it is all circumstantial at best. Better matter will be found in some textbooks on physical chemistry, notably in Nernst's book entitled "Theoretical Chemistry from the Standpoint of Thermodynamics and Avogadro's Rule," where the hypothesis of a thin diffusion layer is applied to many physicochemical processes.

Some of the above data on absorption are taken from Bulletin No. 51 of the United States Bureau of Soils.

The intention of the authors is to put heat transmission, electrolytic diffusion (conduction), and heterogeneous chemical reactions all on the same footing so far as concerns their being functions of relative bodily velocities between the different states of the matter concerned. The authors do not desire to have this velocity factor given overwhelming importance compared with other factors, but to have it given the attention it deserves.

A number of physical chemists have pointed out that many measurements of reaction velocities heretofore made are really measurements of secondary and incidental physical velocities. Possibly a careful and discriminating application of this "scrubbing" factor will serve to harmonize a number of discordant measurements and apparently contradictory facts.

As a contribution in that direction the following abstract treatment is offered bearing on the evaporation of water from its free surface (not from a wetted solid) in a wind. Some physicists ought to investigate the subject with accuracy, because some really well-founded equations on such heterogeneous reactions would be of immense service in physical chemistry, physics, and engineering.

- Let
- p = the pressure of the water vapor in the air.
 - P = the vapor pressure of the water.
 - F = the surface area of the water.
 - D = the diffusion coefficient of the vapor from the water.
 - δ = the thickness of the film of air adhering to the surface of the water.

^a Ann. Phys., ser. 4, vol. 3, 1900, p. 328.

Let x = the weight of water evaporated up to the time t .
 t = time.

V = velocity of the moving air.

a and b = (unknown) constants.

Then the rate of evaporation at end of time t ,

$$\frac{dx}{dt} = DF \frac{P-p}{\delta},$$

which, on disintegration under the condition that $x=0$ when $t=0$, gives $\frac{x}{tF} = D \frac{P-p}{\delta}$.

The quantity $\frac{x}{tF}$ is the amount evaporated from unit surface in unit time E ; then

$$E = D \frac{P-p}{\delta} \quad (\text{I})$$

Now, introduce the experimental fact (see Fitzgerald formula p. 169) that the amount of water evaporated bears a linear relation to the velocity of the wind over the water surface; then the evaporation must be a function of the velocity of the air according to an equation of some such form as the following:

$$E = D(P-p)(a+bV) \quad (\text{II})$$

Equating (I) and (II) gives

$$E = \frac{D(P-p)}{\delta} = D(P-p)(a+bV).$$

Whence

$$(a+bV)\delta = 1$$

In the heat-impartation equations (probably substantiated by the Geological Survey's experiments), when all complicating "other things" are rendered "equal," $a=0$ so that

$$\delta V = \frac{1}{b},$$

and the curve showing the relation between the thickness of the diffusion layer and the velocity of the gas is a rectangular hyperbola. But in evaporation a is not 0, and the hyperbola is distorted by a higher velocity coordinate. By careful experiments it might be feasible to plot this distorted hyperbola in inches (or centimeters), and then a very clear idea could be had of the magnitudes involved. This, the authors believe, would be the first case in which a diffusion layer in gases would have been calculated.

GLOSSARY.

ABSOLUTE TEMPERATURE. The temperature of a substance reckoned from that temperature—461° below the zero on the Fahrenheit scale and 273° below the zero on the centigrade scale—at which all heat is supposed to be absent.

APPROACH, VELOCITY OF. *See* Velocity of approach.

AVAILABLE HEAT. That portion of the heat given to the gases entering a boiler that is above the quantity required to raise the gases from atmospheric to boiler-water temperature.

BAFFLE. A partition designed to change the course of moving gases in the furnace or among the boiler tubes.

BLACK BODY. A term used to designate a body that would absorb absolutely all of any radiant heat (including light) that might fall upon it. Lampblack absorbs about 96 per cent of such radiation. In the laboratory the full equivalent of a perfectly black surface is obtained by using a hollow body of any suitable substance (for example, graphite or earthenware) with a small hole in it, the whole surface being kept uniformly at the temperature under investigation. It is obvious that of any heat entering the hole by radiation, only an extremely small part would ever get out.

BOLTZMANN AND STEFAN'S LAW. *See* Stefan and Boltzmann's law.

BRITISH THERMAL UNIT. The quantity of heat that is required to raise the temperature of 1 pound of pure water from 32° F. to 212° F., divided by 180 (sometimes designated "average British thermal unit"). The abbreviation B. t. u. is used in this bulletin.

CAPACITY. A word commonly used, loosely, to denote the percentage of steam made by a boiler; the rated evaporation of the builder being taken as 100 per cent.

COMBUSTIBLE. When in quotation marks this word means "coal free from moisture and ash."

COMBUSTION CHAMBER. Strictly, the entire space between the surface of the fuel bed and the area at which the gases enter the boiler. The term is generally used, however, to designate the space between the bridge wall and that area.

CONDUCTION. The process of transferring heat by direct contact, as along a rod, or from a hot stove to a flat iron on it.

CONVECTION. The addition of heat to or the removal of heat from a body by gases or liquids circulating in direct contact with the body; as the removal of heat from a radiator by the circulation of air.

CRITICAL VELOCITY. Used, in connection with a moving fluid, to designate that velocity at which the stream lines of flow cease rather suddenly to be parallel, and at which violent eddying begins.

In this bulletin the term is also used in the discussion of the tests on the laboratory multitubular boilers, to designate that velocity of the air entering the boiler tubes beyond which the true boiler efficiency is substantially constant or falls off slightly. This critical velocity is probably a manifestation of the appearance of eddying.

DRY SURFACE. On the gas side of a boiler tube, that thin zone or region in the layer of soot and gas at which it is presumed the heat ceases to travel by radiation and convection and begins to travel by conduction.

EFFICIENCIES. Boiler and grate ("over-all") efficiency, A.S.M.E. code item 73, is the ratio of the heat absorbed by a boiler to the potential heat of the dry coal fired.

Boiler efficiency 72.1 (designated in bulletins of the United States Geological Survey as 72*) is the ratio of the heat absorbed by a boiler to the potential heat of the "combustible" ascending from the grate. The "combustible" in this item is equal to the total dry coal fired, minus the ash shown by analysis of the dry coal, minus the combustible in the refuse.

Percentage of completeness of combustion is the ratio of the heat actually evolved in the furnace to the potential heat of the combustible ascending from the grate.

True boiler efficiency, sometimes denoted as E_4 by the present authors, is the ratio of the heat absorbed by the boiler to the heat in the gases that is available to the boiler, only that part of the heat in the gases being considered available which remains after subtracting from the total heat actually generated the quantity used in heating the gases from atmospheric to the steam temperature of the boiler.

EMPIRICAL FORMULA. A formula expressing the actual relations between two or more variables and constants, but not founded on known laws. *See* "Rational formula."

FLUE-GAS TEMPERATURE. A term used to denote the temperature of gases as they leave the heating plates of a boiler.

FLUES. Synonymous with the tubes of a multitubular (fire-tube) boiler.

HEAT, AVAILABLE. *See* Available heat.

HEATING PLATE. A term used in this bulletin to denote any metallic part of the boiler, one side of which is either exposed to direct radiation from the furnace or is in contact with the hot furnace gases, and the other side of which is wetted by the boiler water. This term is used in preference to the common expression "heating surface," in order to distinguish dry from wet surface.

HORSEPOWER, BOILER. A boiler can not logically be rated in horsepower. Nevertheless, in the United States a boiler horsepower is said to be equivalent to the evaporation of 34.5 pounds of water per hour into steam from and at 212° F.

INITIAL TEMPERATURE. A short term used to denote the temperature of air or other furnace gases entering the boiler.

POTENTIAL HEAT. A term applied to the undeveloped heat in fuel, as determined by a calorimeter.

PRESSURE DROP. The difference between the pressures ("drafts") at two places in a flowing stream of gas.

RADIATION. The process of transferring energy (including heat, light, etc.) through space without the aid of tangible material; for example, from the sun to the earth.

RATIONAL FORMULA. A formula deduced from fundamental laws, as of physics. *See* Empirical formula.

STEFAN AND BOLTZMANN LAW. The net amount of radiant energy (heat, for example) radiated from a black surface to a cooler black surface is proportional to the difference of the fourth powers of the absolute temperatures of the bodies.

TEMPERATURE, ABSOLUTE. *See* Absolute temperature.

TEMPERATURE GRADIENT. As used in this bulletin, any continuously rising or falling change of temperature along or through a body or bodies actively conducting heat.

VELOCITY, CRITICAL. *See* Critical velocity.

VELOCITY OF APPROACH. When water flows from a weir in the end of a large trough the water is said to approach the weir with negligible velocity. But if the trough be small or tapered toward the orifice, then the water approaches the orifice with considerable velocity, so that the discharge is greater than that due merely to the static head on the weir. The same reasoning applies fully to gases.

As a consequence of this effect it often happens, under the same pressure drop ("draft") through two boilers identically alike (speaking of the boilers proper), that much more gas will pass through one than through the other, because the passage immediately preceding the first boiler is of smaller cross section, whence the gas velocity is higher.

WET SURFACE. On the water side of a boiler tube that thin zone or region in the adhering film of water at which, it is presumed, the transmitted heat ceases to travel by conduction and is removed by the convection of the water.

PUBLICATIONS ON FUEL TESTING.

The following publications, except those to which a price is affixed, can be obtained free by applying to the Director of the Bureau of Mines, Washington, D. C. The priced publications can be purchased from the Superintendent of Documents, Government Printing Office, Washington, D. C.

PUBLICATIONS OF THE BUREAU OF MINES.

BULLETIN 1. The volatile matter of coal, by H. C. Porter and F. K. Ovitz. 1910. 56 pp., 1 pl.

BULLETIN 2. North Dakota lignite as a fuel for power-plant boilers, by D. T. Randall and Henry Kreisinger. 1910. 42 pp., 1 pl.

BULLETIN 3. The coke industry of the United States as related to the foundry, by Richard Moldenke. 1910. 32 pp.

BULLETIN 4. Features of producer-gas power-plant development in Europe, by R. H. Fernald. 1910. 27 pp., 4 pls.

BULLETIN 5. Washing and coking tests of coal at Denver, Colo., July 1, 1908, to June 30, 1909, by A. W. Belden, G. R. Delamater, J. W. Groves, and K. M. Way. 1910. 62 pp.

BULLETIN 6. Coals available for the manufacture of illuminating gas, by A. H. White and Perry Barker, compiled and revised by H. M. Wilson. 1911. 77 pp., 4 pls.

BULLETIN 7. Essential factors in the formation of producer gas, by J. K. Clement, L. H. Adams, and C. N. Haskins. 1911. 58 pp., 1 pl.

BULLETIN 8. The flow of heat through furnace walls, by W. T. Ray and Henry Kreisinger. 1911. 32 pp.

BULLETIN 9. Recent development of the producer-gas power plant in the United States, by R. H. Fernald. 82 pp., 2 pls. Reprint of United States Geological Survey Bulletin 416.

BULLETIN 11. The purchase of coal by the Government under specifications, by G. S. Pope. 80 pp. Reprint of United States Geological Survey Bulletin 428.

BULLETIN 12. Apparatus and methods for the sampling and analysis of furnace gases, by J. C. W. Fraser and E. J. Hoffman. 1911. 22 pp.

BULLETIN 13. Résumé of producer-gas investigations, October 1, 1904-June 30, 1910, by R. H. Fernald and C. D. Smith. 1911. 393 pp., 12 pls.

BULLETIN 14. Briquetting tests of lignite, at Pittsburgh, Pa., 1908-9; with a chapter on sulphite-pitch binder, by C. L. Wright. 1911. 64 pp., 11 pls.

BULLETIN 16. The uses of peat for fuel and other purposes, by C. A. Davis. 1911. 214 pp., 1 pl.

BULLETIN 21. The significance of drafts in steam-boiler practice, by W. T. Ray and Henry Kreisinger. 64 pp. Reprint of United States Geological Survey Bulletin 367.

TECHNICAL PAPER 1. The sampling of coal in the mine, by J. A. Holmes. 1911. 18 pp.

TECHNICAL PAPER 2. The escape of gas from coal, by H. C. Porter and F. K. Ovitz. 1911. 14 pp.

TECHNICAL PAPER 3. Specifications for the purchase of fuel oil by the Government, with directions for sampling oil and natural gas, by I. C. Allen. 1911. 13 pp.

TECHNICAL PAPER 5. The constituents of coal soluble in phenol, by J. C. W. Frazer and E. J. Hoffman. 1912. 20 pp.

TECHNICAL PAPER 8. Methods of analyzing coal and coke, by F. M. Stanton and A. C. Fieldner. 1912. 21 pp.

TECHNICAL PAPER 9. The status of the gas producer and of the internal-combustion engine in the utilization of fuels, by R. H. Fernald. 1912. 42 pp.

TECHNICAL PAPER 10. Liquefied products from natural gas; their properties and uses, by I. C. Allen and G. A. Burrell. 1912. 23 pp.

TECHNICAL PAPER 16. Deterioration and spontaneous combustion of coal in storage, by H. C. Porter and F. K. Ovitz. 1912. 16 pp.

PUBLICATIONS OF THE UNITED STATES GEOLOGICAL SURVEY.

[Transferred to the Bureau of Mines.]

PROFESSIONAL PAPER 48. Report on the operations of the coal-testing plant of the United States Geological Survey at the Louisiana Purchase Exposition, St. Louis, Mo., 1904; E. W. Parker, J. A. Holmes, M. R. Campbell, committee in charge. 1906. In three parts. 1492 pp., 13 pls. \$1.50.

BULLETIN 261. Preliminary report on the operations of the coal-testing plant of the United States Geological Survey at the Louisiana Purchase Exposition, St. Louis, Mo., 1904; E. W. Parker, J. A. Holmes, M. R. Campbell, committee in charge. 1905. 172 pp. 10 cents.

BULLETIN 290. Preliminary report on the operations of the fuel-testing plant of the United States Geological Survey at St. Louis, Mo., 1905, by J. A. Holmes. 1906. 240 pp. 20 cents.

BULLETIN 325. A study of four hundred steaming tests, made at the fuel-testing plant, St. Louis, Mo., 1904, 1905, and 1906, by L. P. Breckenridge. 1907. 196 pp. 20 cents.

BULLETIN 332. Report of the United States fuel-testing plant at St. Louis, Mo., January 1, 1906, to June 30, 1907, J. A. Holmes, in charge. 1908. 299 pp. 25 cents.

BULLETIN 336. Washing and coking tests of coal and cupola tests of coke, by Richard Moldenke, A. W. Belden, and G. R. Delamater. 1908. 76 pp. 10 cents.

BULLETIN 343. Binders for coal briquets, by J. E. Mills. 1908. 59 pp.

BULLETIN 362. Mine sampling and chemical analyses of coals tested at the United States fuel-testing plant, Norfolk, Va., in 1907 by J. S. Burrows. 1908. 23 pp. 5 cents.

BULLETIN 363. Comparative tests of run-of-mine and briquetted coal on locomotives, by W. F. M. Goss. 1908. 57 pp.

BULLETIN 366. Tests of coal and briquets as fuel for house-heating boilers, by D. T. Randall. 44 pp., 3 pls.

BULLETIN 368. Washing and coking tests of coal at Denver, Colo., by A. W. Belden, G. R. Delamater, and J. W. Groves. 1909. 54 pp., 2 pls. 10 cents.

BULLETIN 382. The effect of oxygen in coal, by David White. 1909. 74 pp., 3 pls.

BULLETIN 385. Briquetting tests at the United States fuel-testing plant, Norfolk, Va., by C. L. Wright. 1909. 41 pp., 9 pls.

BULLETIN 393. Incidental problems in gas-producer tests, by R. H. Fernald, C. D. Smith, J. K. Clement, and H. A. Grine. 1909. 29 pp.

BULLETIN 403. Comparative tests of run-of-mine and briquetted coal on the torpedo boat *Biddle*, by W. T. Ray and Henry Kreisinger. 1909. 49 pp. 10 cents.

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